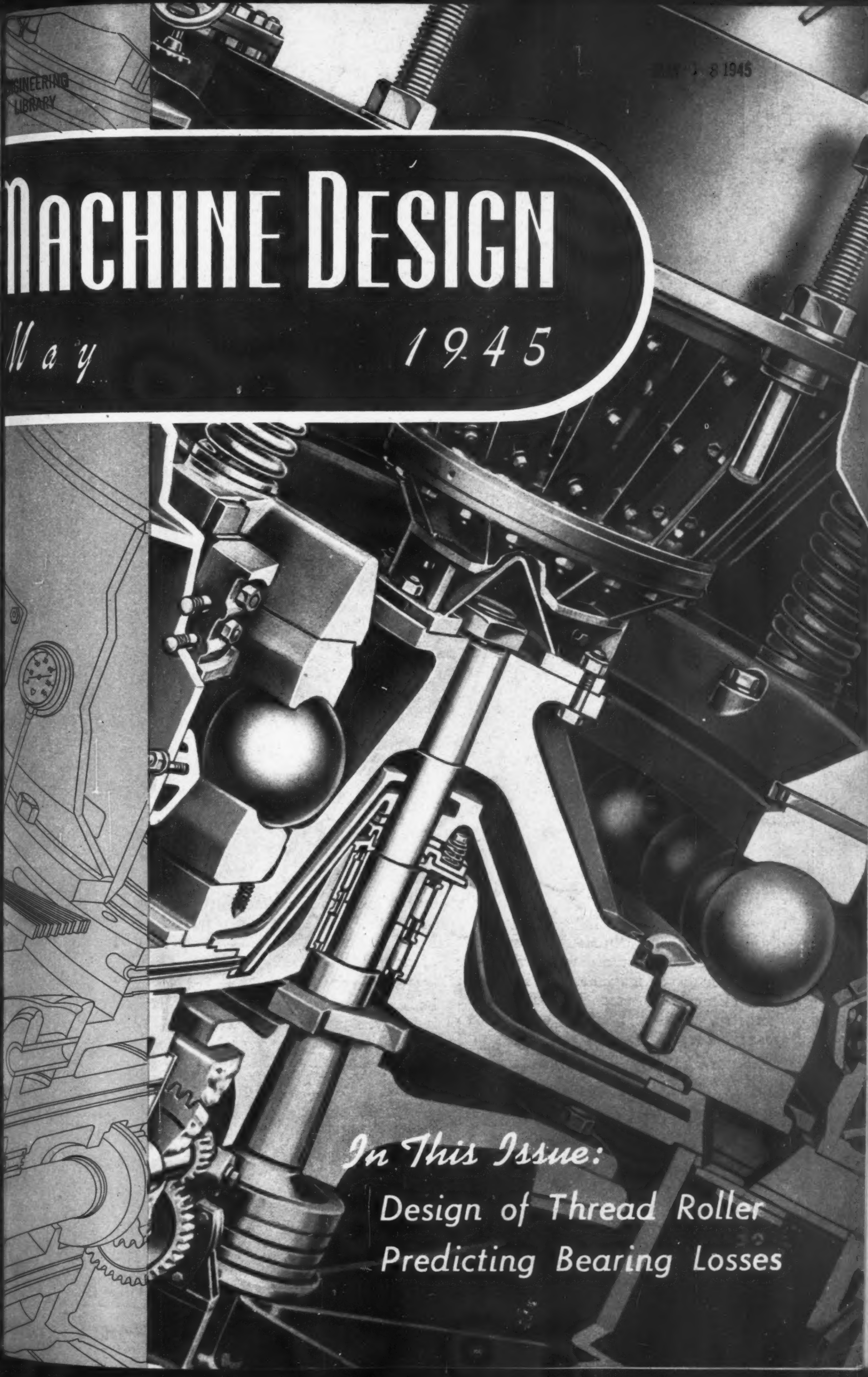
MACHINE DESIGN

May 1945



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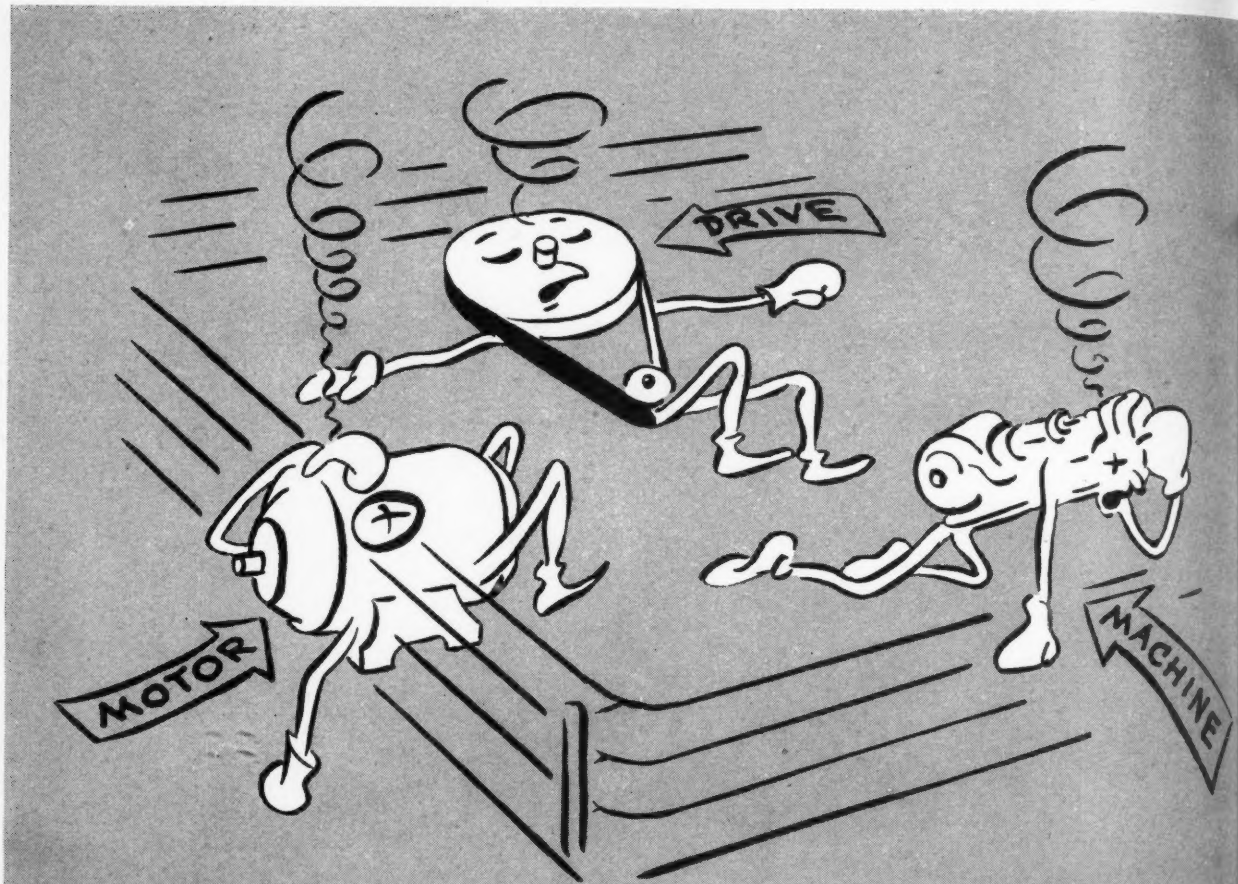
STOP THIS FIGHT



Once Misalignment starts 'em Fighting each other . . . the only Question is: Which will go First?

SPRUNG OR BROKEN shafts, burned-out bearings, overload failure—are cases of motor damage commonly caused by Misalignment. And the damage can occur in drive or driven machine, too. For when these elements are assembled in incorrect geometry, bending, breaking or excessive wear must result. *Something has to "give!"*

Picture above is from Allis-Chalmers' new "Guide to Wartime Care of Electric Motors" . . . which takes a fresh look at Misalignment *and the 8 other main enemies of motor life!*

Over 100,000 copies of this valuable new book are already in use by armed forces and industry. Applies to all makes—contains no advertising. Send for *your* free copy.

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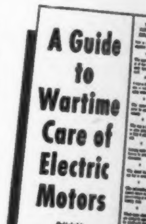

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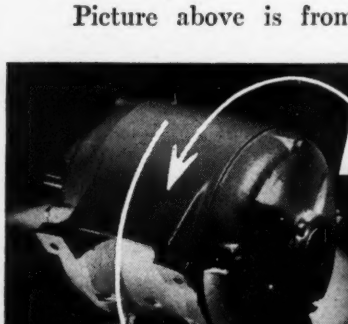


ALLIS-CHALMERS MOTORS

When you do need new motors, look into the strength, solidity and all-around protection of the new "Safety Circle"—protected top, sides, ends and bottom.

A-C

WE WORK FOR VICTORY

WE PLAN FOR PEACE

MACHINE DESIGN

THE PROFESSIONAL JOURNAL OF CHIEF ENGINEERS AND DESIGNERS

Contents

MAY, 1945 Volume 17—Number 5

Better Porcelain Enameled Products from Inland Research

Ti-Namel—The New Alloy Steel Base for Vitreous Enamel Also Lowers Cost of Product

For many years the Inland research staff has been studying and experimenting toward the development of a better base for porcelain enamel—a base that would simplify operations, reduce shop time and labor costs, and produce a superior product. The result of this intensive research is Inland Ti-Namel—the new titanium alloy steel.

During the research period Inland Metallurgists worked on almost every possible combination of alloy. Finally it was determined that titanium would combine with the carbon in the steel to form a sufficiently stable carbide which is essential for the successful application of a thin white cover coat or coats to a base material without the necessity of a ground coat. Then followed a long series of tests to establish the amount of the alloy needed and the process to be used in making this titanium steel. Finally open hearth tests were made and the steel was sent to enameling shops for actual tests in making commercial products. Not until all this preliminary work was completed did Inland announce Ti-Namel—the superior alloy steel base for better porcelain enameled products.

Pending patent applications on the new enameling process and product made thereby are owned jointly by Inland Steel Company and The Titanium Alloy Manufacturing Company under trust agreement.

We have a new descriptive bulletin on Ti-Namel and will be glad to send you a copy.

INLAND STEEL COMPANY

38 South Dearborn St., Chicago 3, Ill.

Sales Offices: Cincinnati • Detroit • Indianapolis • Kansas City • Milwaukee • New York • St. Louis • St. Paul

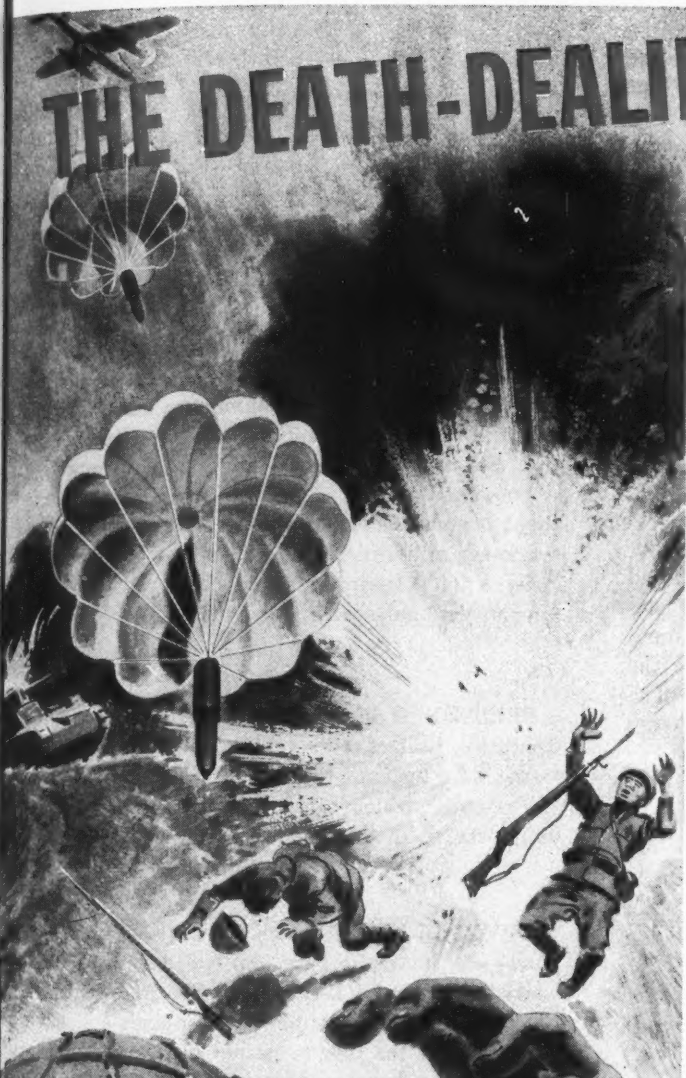
INLAND TI-NAMEL

Itemized Index

Classified for Convenience when Studying Specific Design Problems

MACHINE DESIGN is indexed in Industrial Arts Index and Engineering Index Service, both available in libraries generally.

THE DEATH-DEALING SPRING . . .



THAT DOESN'T SPRING*

IT'S the fragmentation bomb. You'll be hearing more about it as the war progresses. Already millions have been used with devastating effect. Dropped from fast, low-flying bombers, they cut airdromes, encampments and supply trains to pieces. Against ground troops their wide destructive range makes them particularly effective.

Our mills have already produced the casings for more than two million of these bombs—enough to thoroughly saturate Japan if all of them could be dropped in one raid.

Turning out these bombs in enormous quantities required unusual production facilities and the ability to quickly set up streamlined mass production methods that would not sacrifice quality to speed.

It is this ability, plus the unusually high standards of spring engineering that we are able to bring to a job, that has enabled us to produce—by the millions—high precision springs of every type and size, for fighting equipment of every kind.

Certainly the ten-fold increase in our manufacturing facilities plus the things we have learned in these war years about making springs not only faster but better, should be helpful in providing springs for your peacetime products that will be superior both in quality and performance . . . and low in cost as well.

*The dreaded fragmentation bomb consists of a steel cylinder charged with TNT and covered with a spiral-coiled shaped wire which breaks into 1000 to 1500 pieces on explosion. These fragments having velocities up to 4000 feet a second, are highly effective at 200 feet distance.

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U·S·S *American Quality Springs*

Topics

LARGE OPTICAL CRYSTALS, made from sodium nitrate, have been produced by the Polaroid company as large as 7½ x 15 x ¾-inch with optical properties similar to calcite. Prisms cut from these crystals polarize light over a wider range of the spectrum than most other synthetic polarizers. The crystals are formed by floating mica on molten sodium nitrate and gradually cooling.

SAPPHIRES are now being formed from rod shapes into loops to produce thread guides which, because of their extreme smoothness and large radii, cause a minimum of wear on the material being handled.

COLOR FILM, approaching the speed of conventional silver film, will be available within several months and will be an essential adjunct to every research organization, according to predictions of Maj. Perry Thomas, chief of the ATSC Photographic Engineering Branch.

ULTRASMOOTHNESS of external surfaces of the P-80 "Shooting Star" jet fighter is achieved by an air-foil pyroxylin lacquer which is buffed and rubbed to polished-glass smoothness. In addition to the smoothness of this top coat developed by Du Pont, exposure tests indicate it is one of the most durable for this purpose ever developed. A special thinner has been evolved to allow the fast-drying lacquer to flow out to a smooth film.

HIGH-FREQUENCY current picked up from underground wires, supplies power for a vehicle carrying a one-ton load in a Moscow factory, according to the official Soviet News Agency.

ARMOR PIERCING projectiles which have proved so effective in combating the German Panther, Tiger and Royal Tiger tanks have tungsten carbide cores and are much lighter than previous types. This lightness permits a given gun to fire the projectiles with higher velocities, resulting in shorter flight time and more accurate aiming. The core is centered in a housing made mainly of aluminum which streamlines the shell and permits firing from a larger bore gun. For instance, for a 76-mm gun, half the weight is contained in the core, the total weight being about 9 pounds compared with 15 pounds for conventional armor-piercing projectiles. Muzzle velocity is 3400 feet per second compared with 2800 for conventional shells.

PRESSURE SEALING zipper, developed by Goodrich, utilizes overlapping rubber lips applied to the slide fastener to provide an effective seal for any pressure which can be withstood by the structural strength of the fastener.

NEW ALLIGATOR, the LVT-3, has more fire power and heavier armor, and is faster on land and water than the LVT-4. The craft is powered by two 165-horsepower automobile engines and has a ramp door at the rear to permit quick unloading of men and supplies.

INDICATIVE of the importance of powder metallurgy in tungsten products alone, with respect to savings effected, is a calculation made a few years ago. Light then being produced in the United States with tungsten-filament lamps would have cost 3 billion dollars more had lamps with carbon filaments been used.

SENSITIVITY of aircraft instrument bearings is checked in a torque-testing device utilizing a slender wire for the motive power. The slightest "catch" tells the skilled inspector that a microscopic particle is in the bearing.

BEARINGS ARE RETAINED in magnesium alloy housings by utilizing a bushing of 24ST aluminum between the bearing and housing. The bushing ends are spun over the bearing and housing edges, locking the bearing in place. The method, however, is not recommended for bearings having designed thrust loads.

Refining the Design of a Thread Roller

By Richard K. Lotz
Associate Editor, Machine Design

PRECISION in thread rolling has always depended primarily on accurate rolling dies. For some time prior to the war, manufacturers of thread rolling dies had been experimenting with new materials, heat-treating techniques and finishing methods. This resulted in the development of dies sufficiently precise to permit the rolling of even Class 5 threads.

Obtaining accurate thread forms, however, has not been a major problem encountered in thread rolling. Rather, the big problems involved have been due directly to the method whereby the threads were rolled. Until a few years ago, all thread rolling was done between two flat dies as shown in the insert at right of *Fig.* 1. In this process, the blank is placed between the dies and one of the dies is moved lengthwise past the other, rolling the blank

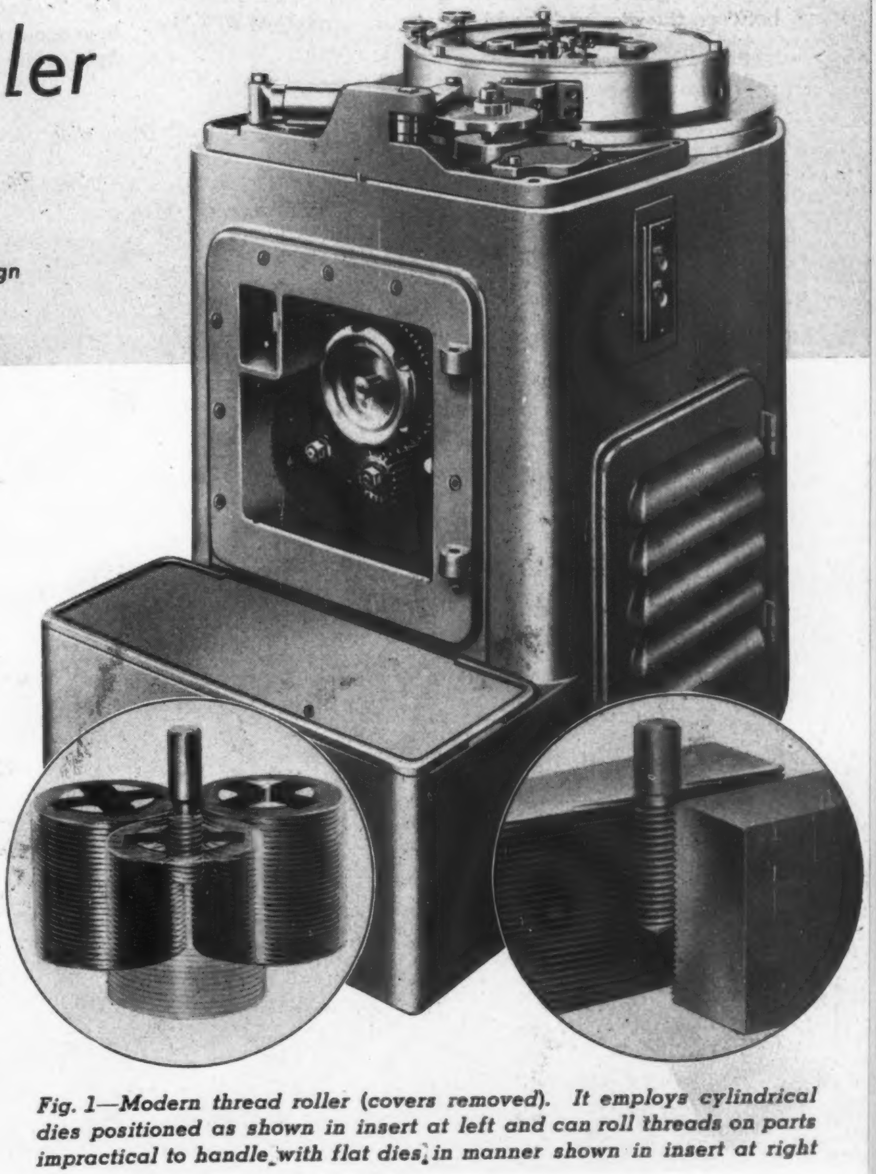


Fig. 1—Modern thread roller (covers removed). It employs cylindrical dies positioned as shown in insert at left and can roll threads on parts impractical to handle with flat dies, in manner shown in insert at right

between them and cold forging the metal into the shape of a thread. The principle involved is identical to that used in all thread rolling, i.e., metal is displaced by cold rolling to conform accurately to the shape of the threads or grooves of the die members.

This flat-die process is subject to some inherent limitations and it was to overcome these and thus broaden the range of application of thread rolling, that engineers of the Rolled Thread Die Company of Worcester, Mass., created a machine utilizing cylindrical dies (see insert at left of *Fig.* 1) in place of the flat dies.

Employment of cylindrical rotating dies makes available, in effect, die surfaces of infinite length, a feature obviously impossible to obtain with flat dies. It is this extension of die surface which obviates one of the primary drawbacks of flat-die rolling—"rapid penetration".

Many Factors Influence Penetration

Broadly stated, as applied to thread rolling, penetration means the rate at which the threads are impressed by the dies into the rolling blank. It will be apparent that the higher the rate is, the greater will be the pressure between the dies and the blank. It is impractical to make flat dies greater in length than 10 to 15 times the circumference, and average penetration (per cont given point on blank with die surface) cannot be than

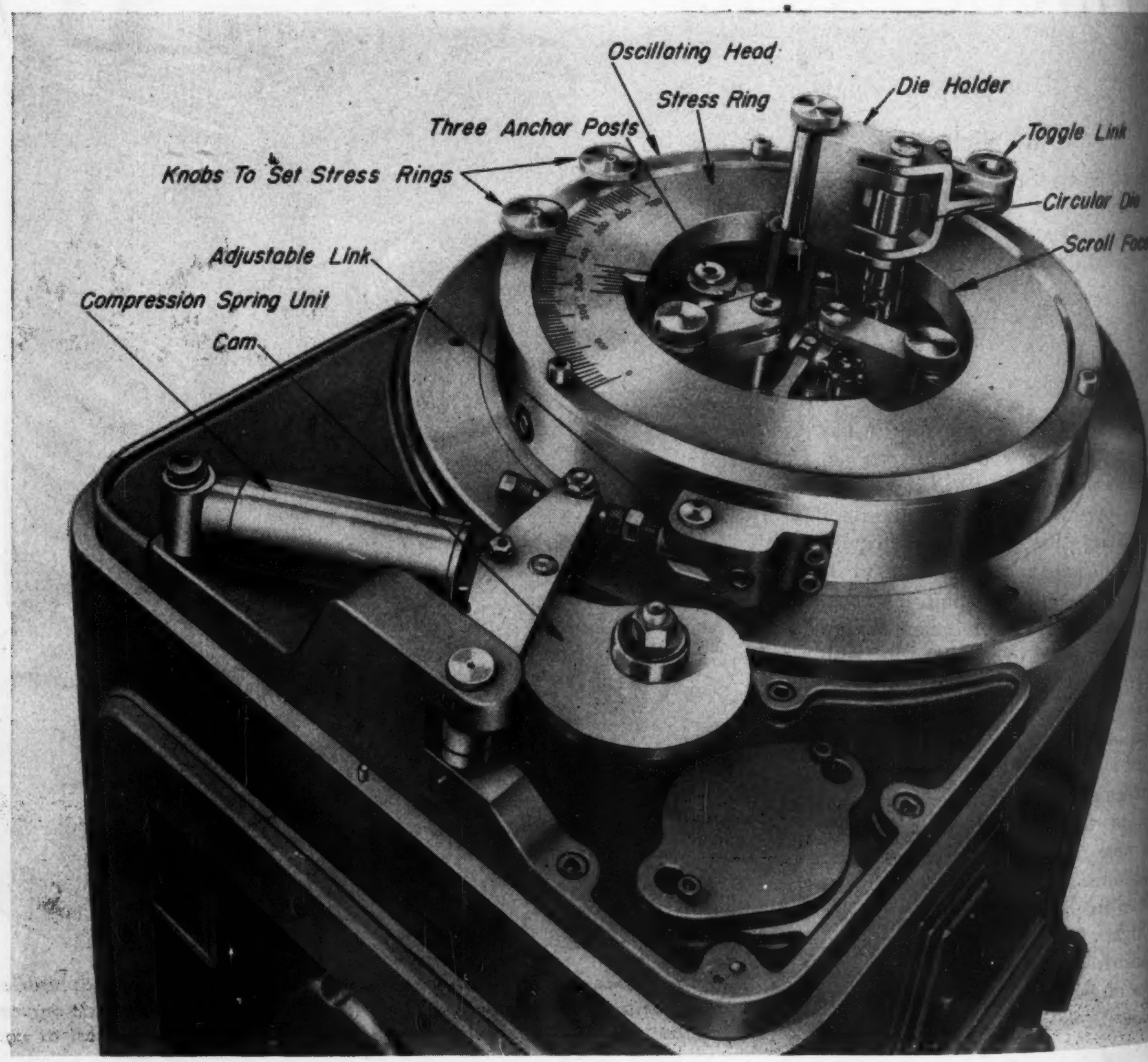
$$\frac{\text{Thread Depth}}{\dfrac{\text{Twice Length of Die}}{\text{Pitch Diam} \times 3.1416}}$$

Thus, for a 1/2-13 thread, rolled on a machine em ing dies the total length of which is 15 inches, ave penetration per contact on the blank is

$$\frac{.05}{\dfrac{2\times7.5}{.45\times3.1416}} = \frac{.05}{10.5} = .0047\text{-inch}$$

Pressures created by such a penetration rate, whi excessive for solid screws and bolts, would collaps distort out of round hollow parts such as pipe fi bushings and spark plug shells, where penetration .001-inch and less per contact often are required. C indrical dies make it possible to effect penetration r

Fig. 2—Top view of machine with cover removed sh how oscillating head is driven by cam through adjust link. Also shows position of toggle joints in stress r



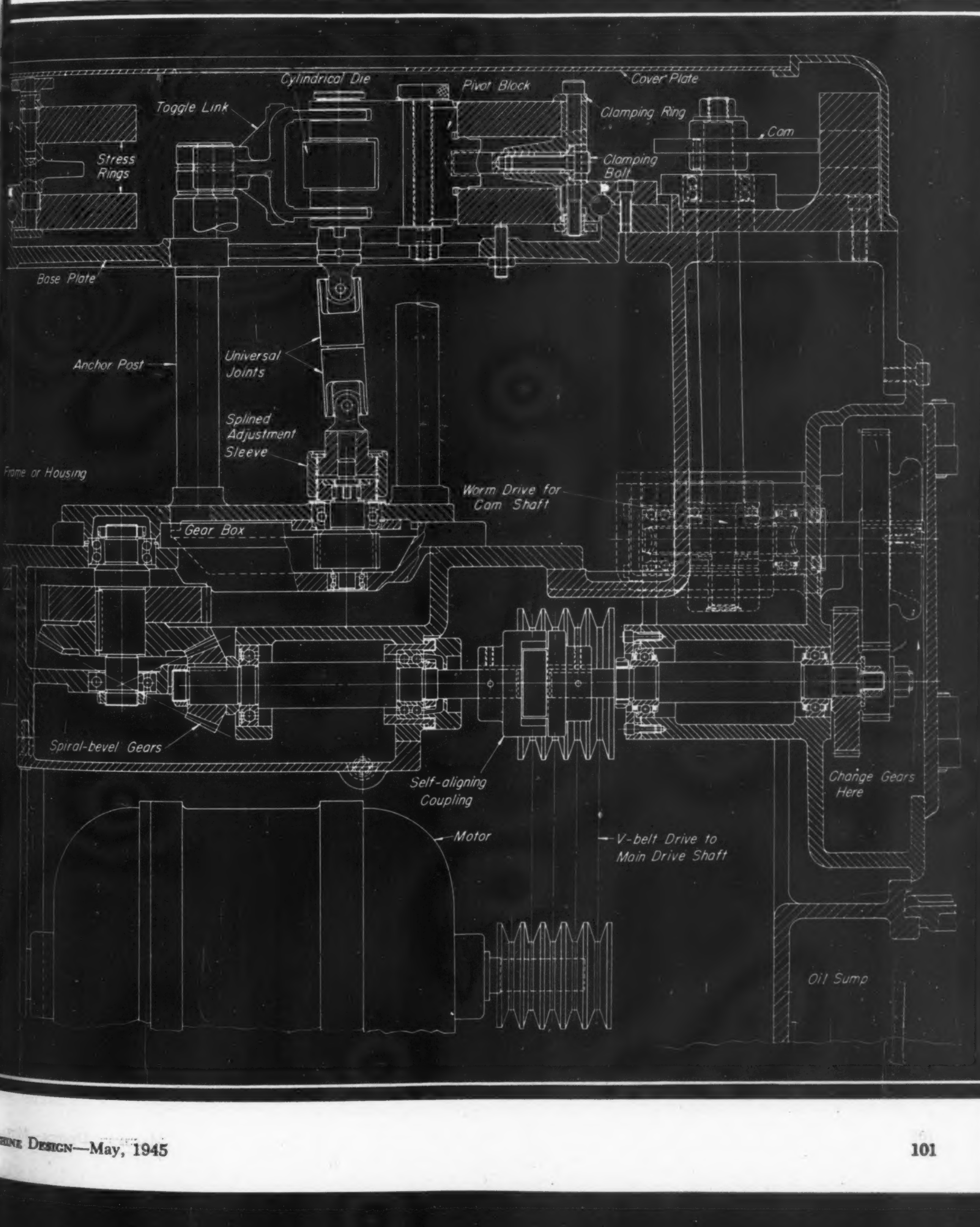
g from extremely low to extremely high.

er limitations of flat-die thread rolling are: Rate of ration can be varied through only a narrow range given cycle to meet the needs of the particular threads shorter in length than the thread diameter enerally impracticable because the blank has a ncy to skew between the dies, spoiling the thread often breaking the dies (this is particularly true there is a long or top-heavy shank above the thread); and finally, because the die lengths are limited, flat-die thread rolling does not generally prove feasible for threads over 1 1/4 inches in diameter.

The machine that has overcome these limitations is pictured in *Fig.* 1. Through proper utilization of cylindrical dies it:

1. Rolls threads on hollow parts by providing a sufficiently low rate of penetration
2. Provides means of varying the penetration rate in a given cycle to handle special threads, materials and shapes of blank
3. Rolls threads shorter in length than is feasible by the

3—Cross section through machine shows main drive motor through change gears, spiral-bevels and spurs spindles. Oscillating head turns on ball bearing



flat-die type of thread rolling machine

4. Rolls threads larger than 1 1/4 inches in diameter.

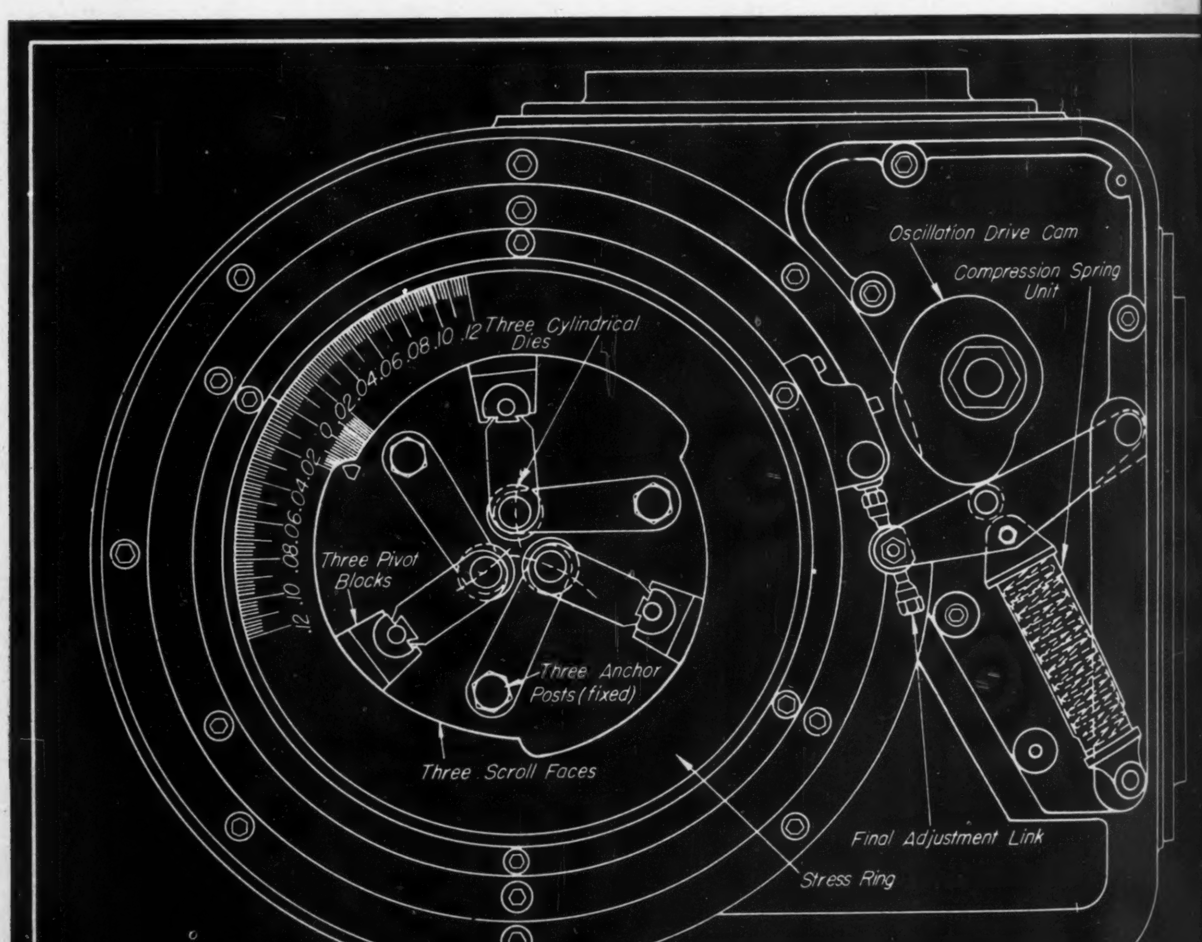
In the head of this machine are located the set of three cylindrical dies positioned as shown in the insert at the left of *Fig.* 1. They are mounted at the apex of a toggle joint (see *Fig.* 2) and driven through universal joints from a gearbox as shown in *Fig.* 3. The drive is taken direct from the motor—located in the base of the machine—via multiple V-belts to the main driveshaft.

To effect the necessary speed reduction between the main driveshaft and the die spindles, the first model of the machine employed a worm drive. However, experience with this drive showed that it did not yield as high an efficiency as was desirable, with the result that in the present machine, speed reduction is effected through a set of spiral-bevel gears as indicated in *Fig.* 3. These have proved to be quiet in operation and high in efficiency. Before driving through the bevels, power is transmitted through a pair of speed-reducing gears. Beyond the bevels is placed a small sun gear, about which are equally spaced the three planet gears that drive the cylindrical dies through novel adjustment couplings and universal joints.

It is generally known, of course, that helical gears are quieter in operation than straight spurs. However, this quietness of operation is only one of the reasons helicals have been used in the change-gear box show the left of the drawing of *Fig.* 3. The large ge mounted on a splined shaft and can be moved o mesh with the main-drive pinion whenever it is de to turn the oscillating cam by hand. Were straight s used for these two gears, it would have been necess hold the large gear in mesh position with a nut o some other clamping means. With helical gears, h ever, it is necessary merely to design the helix ang the teeth to produce the proper direction of end th which then serves to hold the large gear in mesh.

Because it is required to feed the three rotating synchronously into and away from the work, a po ful and positive means had to be developed to move dies in properly timed cycles and at prescribed rat feed. This was accomplished by mounting the t joints, which hold the dies, on a pair of oscillating s rings as shown in *Fig.* 2. Reference to *Fig.* 3 will a clear conception of how the stress-ring unit has designed. The anchor posts, on which are pivoted links of the toggle joints, are fastened into holes of base plate. The two stress rings are rotatable w the outer ring and clamped to the outer ring by mea

Fig. 4—Top view showing how toggle action for m dies into and away from work is obtained. Dotted define paths of dies when stress rings oscillat



two clamping rings and a series of bolts.

...in referring to *Fig.* 2, the die holder members of ...ggle joints are pivoted in the pivot blocks which are ...ed to the stress rings and outer ring by draw bolts. ...n the entire head is oscillated, the pivot blocks ...e with the head, but the pivots of the toggle links ...the anchor posts) remain fixed. Thus the resulting ...le action (see *Fig.* 4) swings the dies into and away ...the work as the head oscillates.

...d it been necessary merely to oscillate the head ...r at a constant speed or perhaps in harmonic motion, ...uld have been possible to use a reciprocating rack ...e or a scotch yoke or eccentric. However, as has ...mentioned in the foregoing, the rate of penetration ...e of moving dies into the work) had to be different ...different jobs. Consequently it was decided that a ...-cam drive, designed to accommodate a variety of ..., would be most logical.

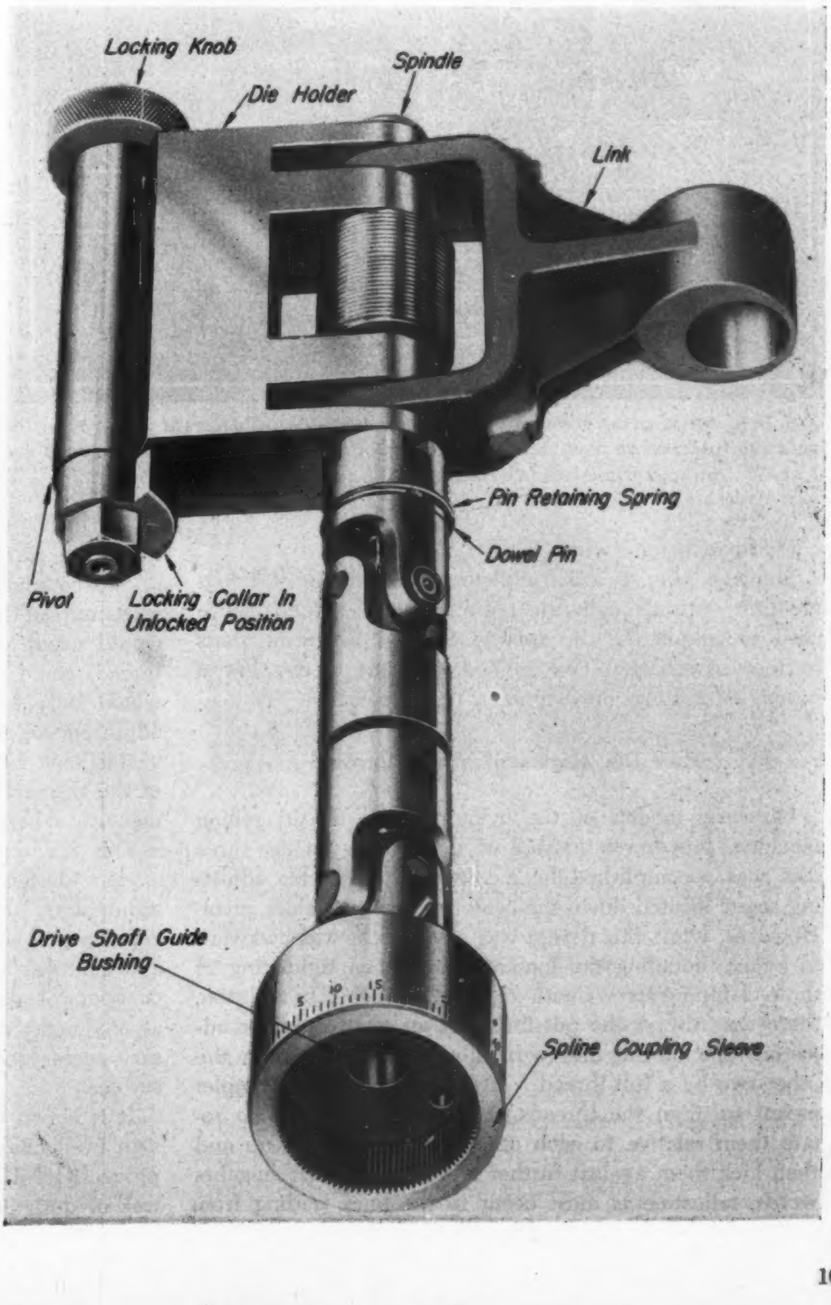
...e plate-cam drive is clearly shown in *Figs.* 2 and 4. ...arm, pivoted on the top plate of the machine, is ...g-loaded against a plate cam and ...s, the head back and forth in oscil...n through an adjustable link. The ...pression spring employed is housed ...telescoping cylinder unit. Employ...t of cams of various contours natural...akes possible oscillation rates of wide ...e, and in combination with change ...s driving the cam at different speeds, ...penetration rates of unlimited variety ...obtainable.

...t will be seen in *Fig.* 2 that the in... diameters of the stress rings are ...ed into scrolls which are noncon...ric with the ring outside diameters. ...e are three such nonconcentric ...lls to which are clamped the pivot ...ks. It is by this means that rough ...stment to the required rolled-thread ...h diameter is made. Clamping the ...t blocks at various points along ...e scrolls results in adjustment of the ...ing between the three cylindrical ...

...his brings into the picture another ...ure recently added to the machine. ...previous models, the link between the ...-driven arm and block mounted on ...outside diameter of the oscillating ...d was of fixed length. Thus, all ...stment of the cylindrical dies to ...l, precise pitch diameter had to be ...ted by shifting the pivot blocks ...g the scroll faces of the stress rings. ...le this procedure produced the de...d results, it nevertheless was some...t awkward, requiring unclamping ...reclamping of both stress rings and ...pivot blocks.

Final adjustment in the present model is made more quickly and easily through the use of an adjustable link between the driving arm and oscillating head. Like all good design it is quite simple, being substantially a bolt, adjustable through the center block by means of an adjusting nut and a checknut.

Exemplification of how one design improvement can lead to others is afforded by consideration of the influence of the adjustable link on the design and use of the scroll faces. As has been mentioned, before development of the adjustable link, final adjustment of the cylindrical dies had to be effected solely by movement of the pivot blocks along the scroll faces. To make precise adjustment by this means required that shallow depths of scroll be employed and this in turn meant that spacing blocks be used between the pivot blocks and scroll faces when rolling small diameter threads. With addition of the adjustable link, coarser preliminary adjustment with deeper scrolls could be tolerated and, in addition, the spacing blocks which formerly were required



...5—Right—Assembly shows how ...drical die is mounted at apex of ...gle formed by die holder and link

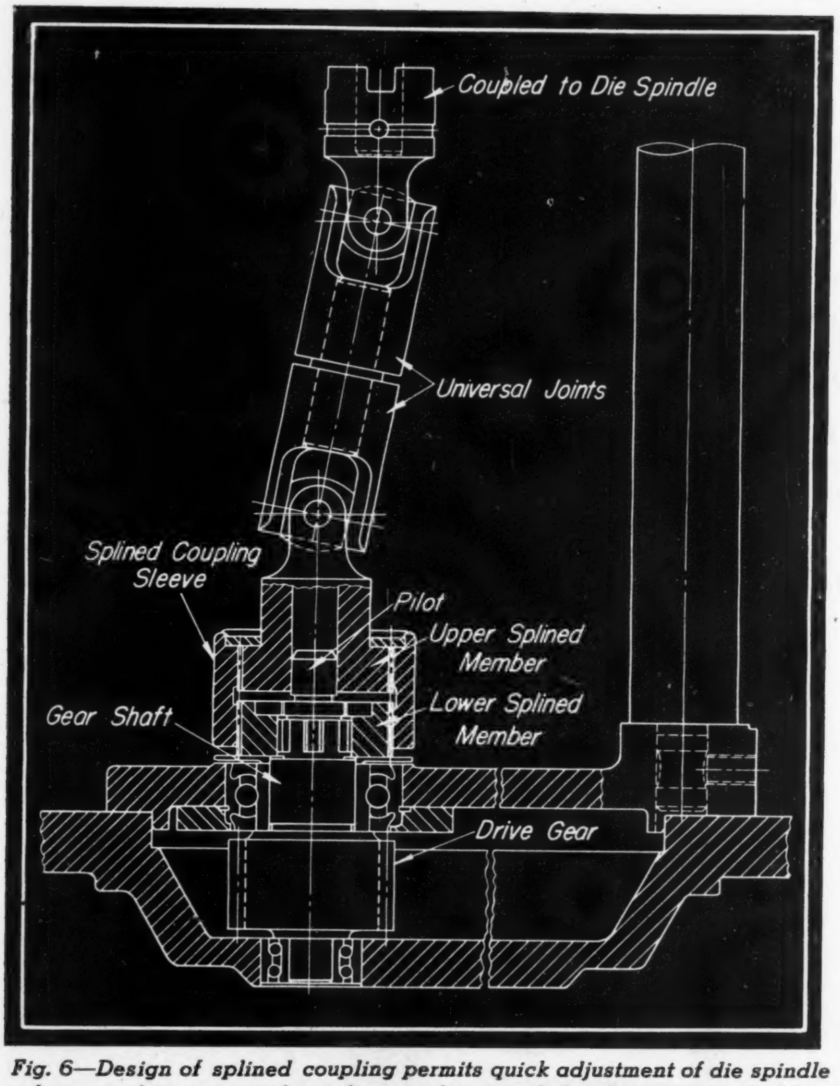


Fig. 6—Design of splined coupling permits quick adjustment of die spindle relative to drive gear without the use of tools. Splined sleeve is lifted by hand, turned, and then dropped back into position shown

could be dispensed with.

Shown in *Fig.* 5 is a complete unit assembly of a toggle unit with its cylindrical die and the splined sleeve used to couple the die spindle through universal joints to the driveshaft. The splined coupling sleeve has a rather interesting development background.

How Die Alignment Was Improved

On early models of the cylindrical-die thread rolling machine, proper relationship of the threads on the three dies was accomplished by a vertical micrometer adjusting screw located down the center of the die-holder pivot. However, while this design was effective, it was awkward to adjust, entailing the loosening and then tightening of three clamping screws with an offset wrench. In addition, there was always the possibility of an inept operator adjusting one of the dies vertically out of level with the other two by a full thread. It was decided that a simpler means to align the threads of the dies would be to rotate them relative to each other until they lined up and then lock them against further relative rotation. In other words, adjustments must occur in the lines leading from the driving gears to the spindles o[f] cylindrical dies.

There are many designs that m[ight] have been used to achieve this ad[just]ment. For example, perhaps the [most] obvious would be to use set sc[rews.] Again, a method using jam nuts m[ight] have been adopted. Still furthe[r, a] clamping coupling could have bee[n de]veloped. All of these, however, w[ould] have been crude and inconvenient [com]pared to the design shown in Fig[s. 5] and 6.

Fastened to the splined shaft of [the] small driving gear is a splined cou[pling] member having 101 teeth cut o[n its] periphery. Directly above it is an[other] externally toothed member wh[ich] couples to the lower universal joint [and] pilots on a stud fitted into the to[p of] the gear shaft. Enveloping both [ex]ternally splined members is an inter[nally] splined sleeve which can be lifted [out] of engagement with the lower mem[ber,] rotated the required amount and [then] dropped back into mesh. Numb[ered] graduation marks are stamped into [the] sleeve periphery and a pointer is [pro]vided as a fixed reference.

Prime Number of Teeth Used

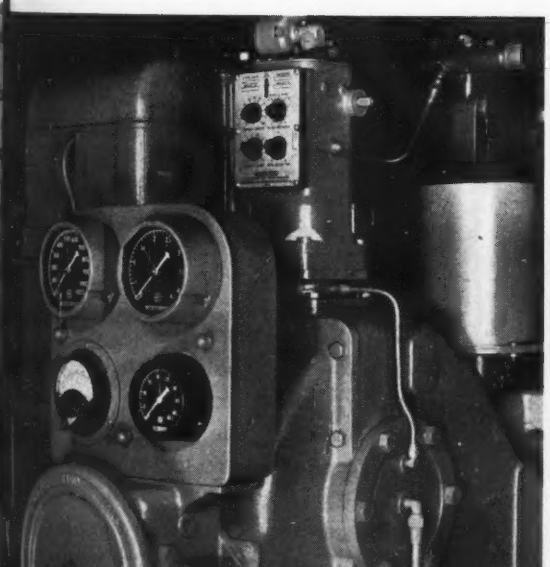
Perhaps the reader is puzzled a[s to] why 101 teeth are used in the cou[pling] sleeve instead of perhaps 100 or [...] The reasoning involved here reverts [...] to the nature of the cylindrical dies [em]ployed. Some of them have s[ingle] threads others have double, triple, quadruple threads and so on up. If the number of [teeth] in the coupling sleeve were divisible by the numbe[r of] starts on the die thread (such as 2, 3, 4, 5, 6, etc.) [there] would occur times when accurate alignment of the threads could not be achieved because turning the sl[eeve] would only keep repeating an inadequate accuracy [of] adjustment. However, since 101 is a prime number [(di]visible only by the number 1 and itself) proper rot[ation] of the sleeve cannot help but wipe out the error in a[lign]ment to a high degree of accuracy.

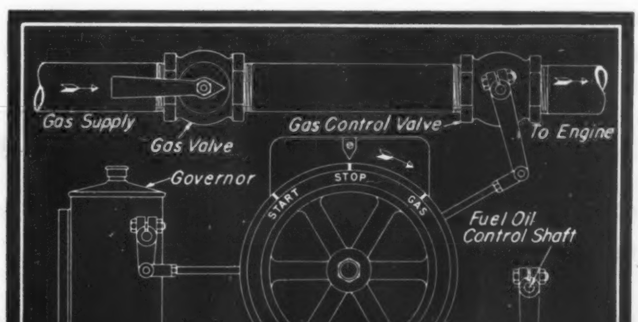
This machine is unquestionably representative of [...] modern design both from an appearance and functi[onal] standpoint. It is extremely compact and streamline[d in] appearance, yet in the accomplishment of these d[esir]able features nothing has been sacrificed in perform[ance] or operator convenience. No projecting knobs ap[pear] at any point on the outside of the machine, the ne[ces]sary access doors and panels being set-in flush with [the] surface.

It is hoped that this discussion of how the various [de]sign problems involved in this machine were solved [will] prove helpful to the reader as he goes about his d[aily] task of designing the ever better machines of the fut[ure.]

Scanning THE FIELD for Ideas

DUAL-FUEL control mechanism now permits a diesel to operate on gas or oil, or both, without any electric sparking device or change in engine parts. Fuel adjustments are effected simply by positioning the control wheel shown in the illustrations. The system, utilizing pilot-oil ignition, was developed by Ralph L. Boyer, chief engineer of The Cooper Bessemer Corp. With this arrangement it is possible to vary the fuel used, without shutdown, in accordance with economic considerations of cost or supply available. The diesel may operate on a wide variety of fuels including fuel oil, natural gas, manufactured and coke-oven gases, sewage gas and refinery by-products.





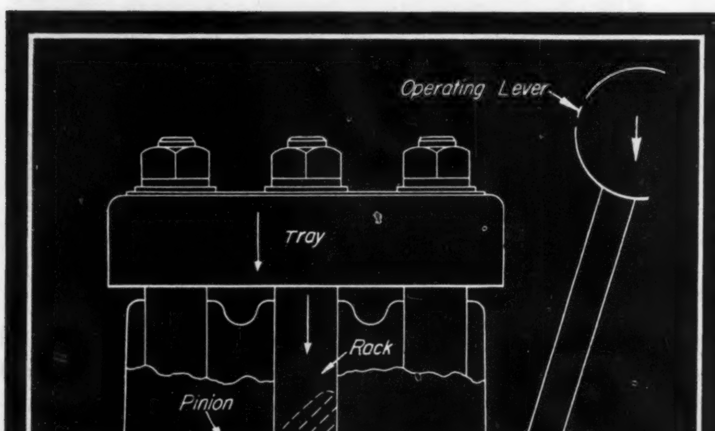
Gas may be introduced under governor control into the intake air stream of an otherwise full-diesel oil engine. This had not been done previously because it was thought that gas would fire on the compression stroke due to the heat of compression. Actually it does not fire because, due to the inherent diesel efficiency, the amount of gas introduced is sufficiently low that the mixture is outside the limits of flammability and will not fire. But, when the pilot oil injection takes place, sufficient heat energy

is supplied to set off the mixture regardless of the percentage. In other words, firing is just as regular at no load as at full load. The pilot charge may be from the standard fuel injection system, except that it is reduced just as it would be for no load condition.

For a 2-cycle engine gas is injected with a timed valve instead of being admitted direct into the intake air as in the 4-cycle system. Operation of diesels as gas engines cuts fuel consumption by as much as 25 per cent over full gas engines.

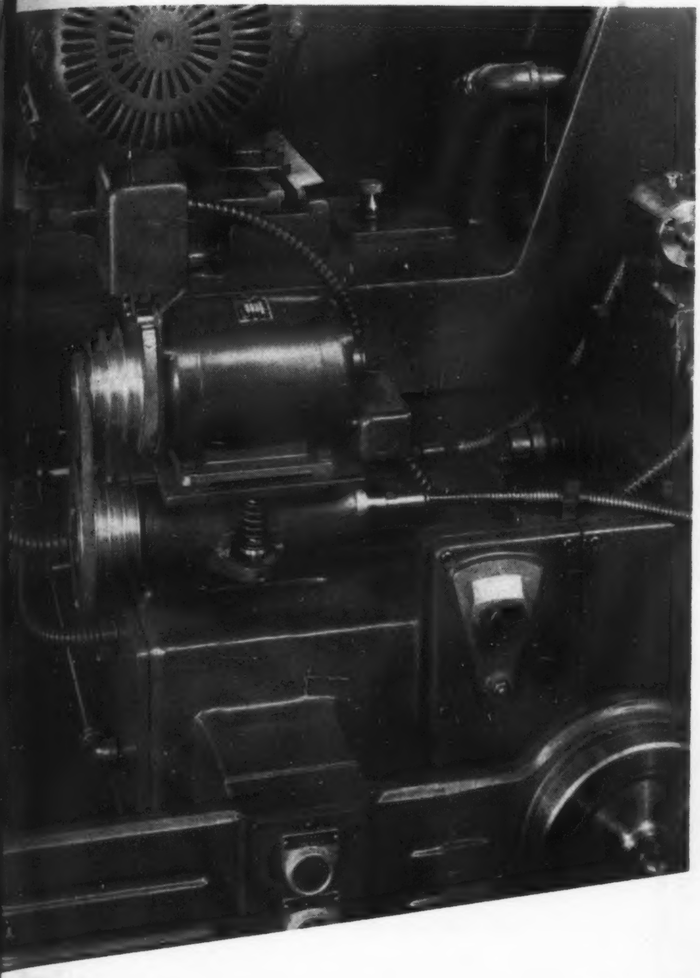
Fast charging of aviators' oxygen cylinders is achieved with a compressor, shown at right, which permits the recharging of more cylinders at a faster rate than previous designs. Developed for the Navy by Walter Kidde & Co. the unit utilizes a conventional crank and crankshaft mechanism to drive the plungers of a 2-stage, water-lubricated, single-acting compresser. Each compression chamber, machined from bronze rod, is provided with poppet type intake and disk type outlet valves. The design permits removal of the compression chamber for cleaning the valves or changing the packing without disturbing any other part. Valve manifolds provide individual control of the flow of oxygen to and from the cylinders, each with master shut-off valves and a pressure gage of the safety-front type, so arranged that in case of accidental rupture the back rather than the glass will blow out.

Charging is effected by partially fi the receiver cylinders by means of boo cylinders, then "topping off" by stor cylinders that are charged automatic at 2200 pounds per square inch. A w trap and two drying towers are provid One dryer removes excessive mois from the oxygen passing through compressor and the second, with tw the capacity, removes excessive moist from the oxygen as it comes from booster cylinders. A frangible type sa ty disk prevents the building up of cessive pressures.



Double-acting lock is simply positively effected through the thrust tion of a helical rack and gear on fit cones on each side of the gear in the illustrated in the sketch at left, design by N. A. Woodworth Co. Locking eff may be obtained on either the up or do stroke as indicated by the force lines the drawing. Pressure on the operat lever rotates the pinion, moving the ra and tray up or down. When the tray co tacts the work, end thrust is develop by the helical pinion and gear, thus pu ing one of the cones against its intern cone. Locking action in this simple in versible system is easily broken by mov ment of the handle.

Miniature Motors Pack a Wallop!



By Fred L. Olson
Motor Application Engineer
Bodine Electric Co.
Chicago

UNPRECEDENTED demands of our armed forces for more and better small fractional horsepower motors have resulted in developments which hieved higher horsepower output per size and weight, proved starting and reversing characteristics and satis-tory operation in extremely wide ambient temperatures mmonly −50 to +55 degrees Cent.) in all kinds of ather conditions from the arctic regions to the tropics. With the anticipated return of increasing civilian pro-ction, improved miniature motors will power new ma-ines and better the performance of standard models. all, dependable motors with or without integral speed ducers will be carefully matched to load characteristics

g. 1—Above—Polyphase motor with external solenoid brake, driving a wheel dresser on a thread grinder

as component parts in machine assemblies. One fractional horsepower motor manufacturer alone offers more than 2500 different motor and gear combinations in compact built-in speed reducer motors available in practically every type of winding from constant-speed synchronous to variable-speed series windings — with complete interchangeability between the alternating current and direct-current wound motors.

The ultimate maximum horsepower rating which meets the required starting torque in a given frame size is largely determined by brake tests in the engineering laboratory (preferably driving the actual equipment), a heat run consisting of thermocouple measurement of the temperature rise on the hottest point of the winding, plus the maximum specified ambient temperature range within the required duty cycle (time on and off). Sometimes a slightly longer or the next larger diameter stacking is recommended to satisfy the operating cycle completely and safely.

Compact, dependable and efficient small motors require careful mechanical construction. Internal friction losses must be kept at a minimum so as not to consume too much of the watts input. Close limits must be maintained on bearing surfaces, shaft end play, preloading of ball bearings, brush clearance and brush spring tension. Likewise, concentricity of end bells with center ring and accurately ground stator bore must exist to guarantee uniform, close air gaps. Furthermore, rotors or armatures running at speeds of 1800 revolutions per minute and higher should be dynamically balanced.

Diamond-bored sleeve bearings are the quietest when tolerances are held to .0002-inch. Nevertheless, extremely wide operating temperatures and installations inaccessible to frequent oiling have necessitated grease-packed ball bearings, long a standard in the machine tool industry, *Fig.* 1. Today's increased ball bearing output with closer limits between balls and races is accelerating the marked

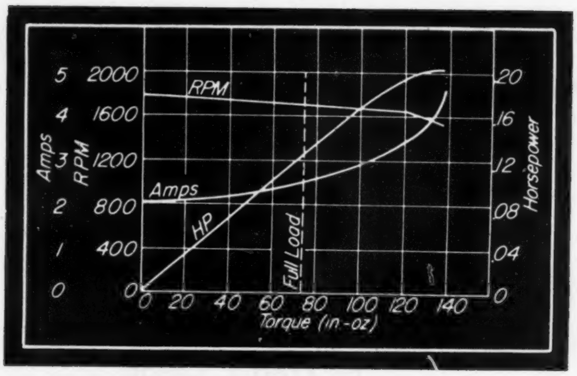


Fig. 2—Split-phase induction motor characteristic curves for a 1/8-horsepower, 1725 rpm motor

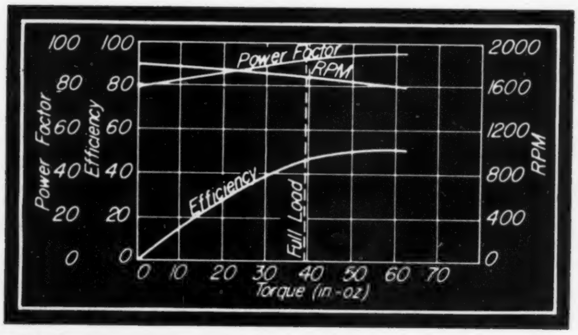


Fig. 3—Characteristic curves for a 1/15-horsepower capacitor motor

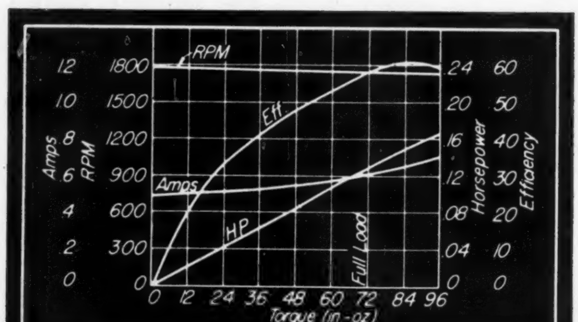
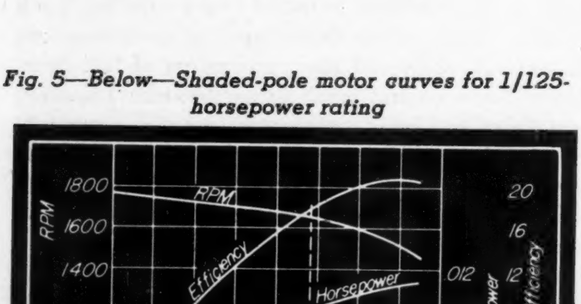


Fig. 4—Curves for a 3-phase, 1725-rpm, 1/8-horsepower induction motor

Fig. 5—Below—Shaded-pole motor curves for 1/125-horsepower rating



trend toward standardizing on ball bearing miniature motors and speed reducers, further reducing friction losses.

Insufficient or inaccurate exchange of performance data between the machine designer and the motor application engineer may lead to motor troubles which are easily avoidable. The information required usually includes the following:

1. Type of machine
2. General description and use
3. Estimated volume of production
4. Voltage, frequency and phase
5. Constant or variable speed
6. Reversible or unidirectional
7. Reversals per minute
8. Running load
9. Starting load
10. Duty cycle (time on and off)
11. Size and weight limits
12. Degree of enclosure
13. Ambient temperature range
14. Surrounding air conditions
15. Mounting position
16. Shaft end-play limits
17. End thrust and overhang loads
18. Dimensions of shaft desired
19. Degree of quietness required
20. Position and length of terminal leads
21. Possibilities of machine stalling.

The foregoing data are not necessarily listed in order of importance. No known or desired operating characteristic should be omitted. For example, caustic air conditions (sometimes present in relatively high humidity) can damage standard windings, injure brushes, corrode shafts, etc., unless special protective treatment or totally enclosing is considered. Degree of quietness needed is best determined by past experience on similar applications and by actual tests in both the laboratory and the field. It should be remembered that "quiet operation" is comparative in degree.

Motor Should Be Protected Against Overload

An inexpensive thermal overload protector is a "must" on many split-phase and some polyphase motors applied to machines which can load up to the point of stalling for even a few seconds at a time. It is too late to prevent a burnt-out or carbonized stator winding when the operator notices the motor smoking on the line! Where infrequent momentary overloads are anticipated—as on a compressor—automatic over-load protection is popular, while on machine tools where abnormal operating load conditions can lock the motor, the manual reset type insures investigation of the trouble. Another (too-often forgotten) way to protect the miniature motor is through installing accurately-rated small fuses, available in compact sizes with suitable mountings.

In some cases the machine designer knows exactly what type and rating of motor to specify from previous experience or tests on similar applications. Often, however, there may be a better and less costly solution by a simpler control connection, or an easier installation by a direct drive through an inexpensive mechanical modification of the shaft or end shield. Also, flange or resilient mounting simplify many applications. Where quantities warrant, a special base, center ring, or unimount face-type bracket can be developed at proportionately low unit cost.

General motor characteristics of various types of minia- motors are shown in TABLE I and are discussed in detail e following.

SPLIT-PHASE motors are the most widely used of all e-phase fractional motors, especially in the smaller rat- . Being one of the simplest to build the cost is low. It ependable because of the good centrifugal switch designs lable today. No slip rings or brush-shifting devices are led since the phase difference between the starting and windings creates the rotating magnetic field. Coming kly up to speed, the pull-up torque permits cutting out high resistance starting winding in a fraction of a second. speed is reasonably constant under slightly varying loads, the starting torque is around 150 per cent of full-load ue. Efficiency and horsepower capacity per frame size both good. Typical performance curves are shown in . 2.

Limitations of Split-Phase Motors

High starting current (five to eight times running cur-) is the most restricting limitation of split-phase motors use of the serious effect on the centrifugal switch and ting winding if started too frequently. They are re- ible only at standstill and, naturally, are not suitable for iable-speed control. Split-phase motors are used exten- ly on washing machines, automatic coin phonographs, rifugal pumps, blowers and machine tools.

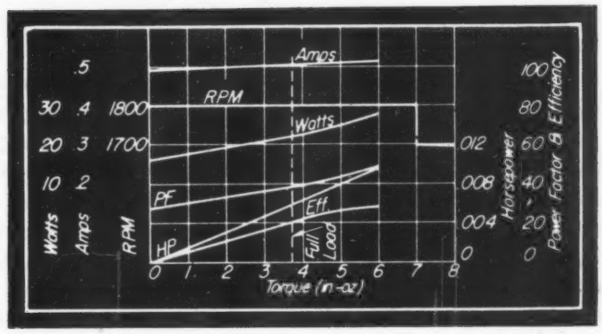
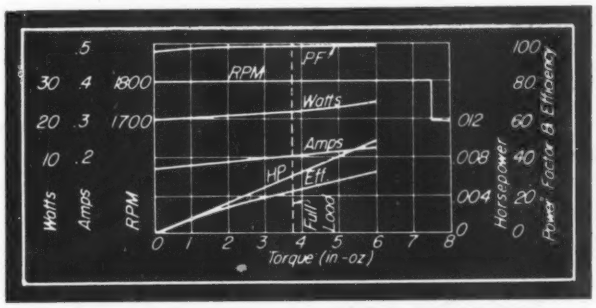


Fig. 6—Above—Synchronous split-phase motor curves for a 1/150-horsepower, 1800-rpm motor

Fig. 7—Below—Synchronous-capacitor motor curves for 1/150-horsepower, 1800-rpm motor



CAPACITOR START-INDUCTION RUN motors are sim- to split-phase motors with an inexpensive electro- condenser in series with the starting winding. added condenser not only increases the start- torque to 300 per cent or more of full load but cuts the starting current as much as 50 per cent. h motors are found on many compressors, vac- n pumps, filing and sawing machines, and other lications having high inertia, static friction and d pressure loads to start. This type also may be d where several starts per minute are a continu- requirement.

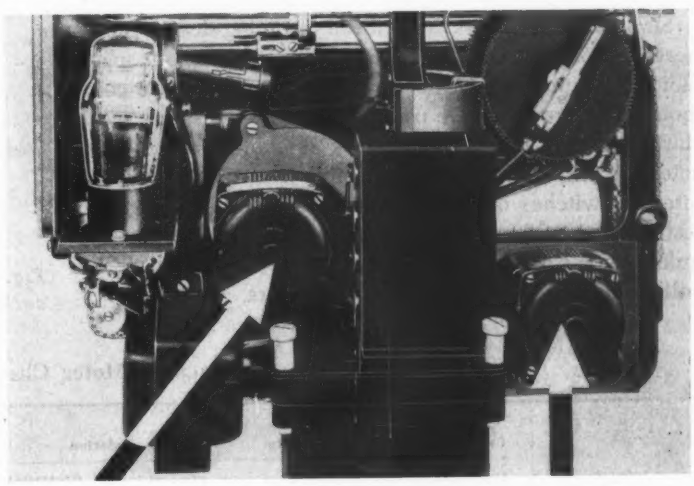
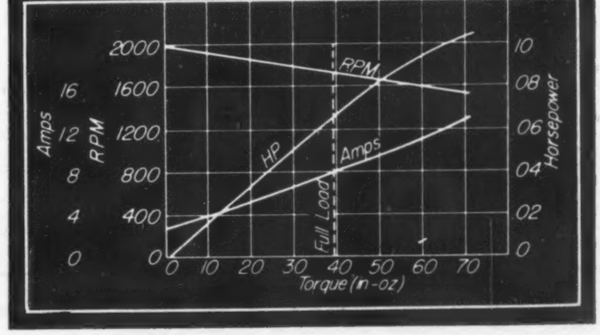


Fig. 8—Above—Pyrometer utilizes a 1/2000-horsepower synchronous motor and a 1/1500-horsepower reversible dynamic braking motor

Fig. 9—Below—Typical shunt-wound motor curves for a 1/15-horsepower, 1725-rpm motor



CAPACITOR START AND RUN motors (commonly erred to as capacitor motors) are being adopted re every year because of their inherent quietness, edom from magnetic hum, dependability, and in- ntaneously reversible feature except on high rtia loads. Either a 3-wire reversible winding h a single-pole double-throw switch, or 4-wire using a ble-pole double-throw switch are common on machine ls, blowers, air conditioners and control apparatus. In the wire type, the capacitor and the main winding are identical d balanced—usually requiring greater capacitance than the ire. No centrifugal cutout switch is required, making it ssible to wind a motor for 60 reversals per minute, depend- g on the size of the motor and the driven machine. The pacitor can be mounted on the motor frame or in any con- nient location on the machine itself.

The capacitor motor has two windings with a capacitor manently in series with one. This creates a phase differ- e, producing a rotating field, starting and running the mo- r much like a two-phase type. Since the starting torque of apacitor motor is usually less than full-load torque, this e is not suitable for high-torque starting. On the other

hand, its high power factor is a desirable feature. Characteristic curves are shown in *Fig.* 3.

POLYPHASE motors in the small, fractional horsepower sizes are sometimes overlooked by the machine designer, yet they develop the highest output per frame size, weight and speed. They should be even more popular in motorizing constant-speed industrial equipment wherever 2-phase or 3-phase power is available. Polyphase alternating current produces a revolving magnetic field in the stator which causes the "squirrel-cage" rotor to revolve at a full-load speed, depending on the frequency and number of poles. Performance curves for a 3-phase, 1/8-horsepower motor are included in *Fig.* 4. Polyphase motors have the following advantages:

1. Simple and dependable construction without centrifugal switch, commutator or brushes
2. Constant speed with no racing at no load
3. Starting torques from 200 to 350 per cent
4. Easily and instantaneously reversible
5. Minimum maintenance.

Close-up view in *Fig.* 1 illustrates a ball bearing type, enclosed, compact polyphase motor with external solenoid brake, driving a diamond wheel dresser of a precision thread grinder. Some other machine applications are hydraulic pumps, coolant pumps, lathes, lens polishers, etc.

Shaded-Pole Motors Are Useful in Small Sizes

SHADED-POLE motors are utilized in timing devices, fans, heaters, and instruments requiring small, constant-speed motors. Ratings of 1/250 to 1/100-horsepower are common. Larger ratings are used less, due to the rather weak starting torque and low efficiency as shown in the characteristic curves, *Fig.* 5. The simple construction without internal switches or brushes plus the favorable magnetic distribution makes this motor popular where extreme quietness and relatively constant speed (affected little by voltage fluctuations) are important factors.

Shaded-pole motors may be stalled for extended pe because of the low starting current—making them sui for torque motor applications of light duty such as matic dampers—but care must be taken to insure adeq starting torque under maximum load. In other words, driven device should require relatively little power, pecially at start. From *Fig.* 5 it will be noted that highest efficiency and output is obtained only when motor is operated at a point where the speed begins fall off appreciably with increased load.

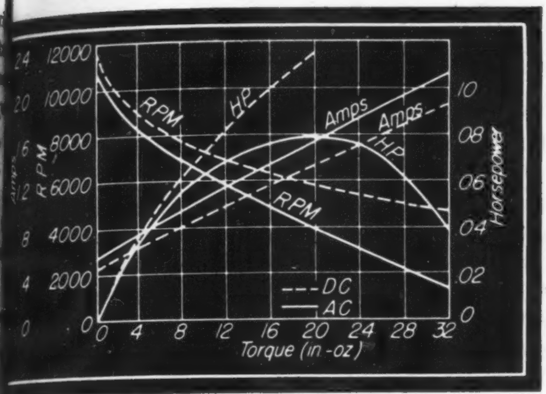
SYNCHRONOUS motors are available in three types windings: Split-phase, capacitor and polyphase. most alternating-current motors they employ a squi cage rotor and distributed stator. Starting as an induc motor, the salient-pole rotor pulls the motor into step

Fig. 10—Portable sander driven by a small, high-spe series motor. Special flange mounting fits sander housi

TABLE I—Motor Characteristics

Motor Type	Current Supply: A-C	Current Supply: D-C	Current Supply: A-C or D-C	Duty: Continuous	Duty: Intermittent	Rotation: Unidirectional	Rotation: Reversible: At rest only	Rotation: Reversible: At rest or rotating	Speed: Constant Fixed	Speed: Constant Adjust.	Speed: Variable	Starting Torque: Low	Starting Torque: Normal	Starting Torque: High	Starting Current: Low	Starting Current: Normal
Split-Phase	X			X			X		X				X			X
Shunt or Compound		X		X				X		X			X			X
Series			X		X	X					X		X*			X
Polyphase	X			X				X	X					X		X
Synchronous	X			X			X		X				X			X
Synchronous Polyphase	X			X				X	X					X		X
Synchronous Capacitor	X			X			X		X			X			X	
Shaded Pole	X			X		X			X			X			X	
Series Governor			X		X	X				X				X		X
Capacitor	X			X				X	X			X			X	
Capacitor Start	X			X			X		X					X		X†

*Starting torque high for series motors with normal speed ratings of 7500 revolutions or more.
†Starting current appreciably lower than for split-phase motors.



11—*Universal series-wound motor performance curves for a 1/15-horsepower, 6500-rpm motor*

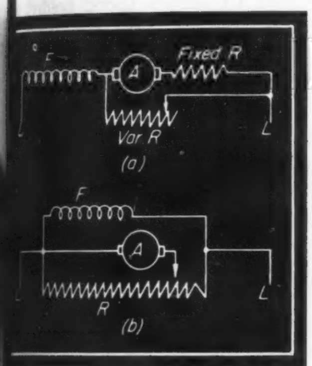


Fig. 12—Diagram for a wide-range speed control which retains stability of a series motor at low speeds is shown at (a). At (b) is a schematic for reducing speed of a small shunt motor without sacrificing good starting performance

roaches synchronous speed. This absolutely constant ed is determined by the number of salient poles and frequency, and will remain constant at loads within range of the minimum pull-out torque.

n interesting comparison is shown in the performance ves of a synchronous split-phase, *Fig.* 6, and a synonous capacitor motor, *Fig.* 7, both of identical horseer ratings. Starting torque and current of the synonous split-phase motor is much greater, but the power tor of the synchronous capacitor type is almost unity.

While the split-phase synchronous motor is the most ely used because of lower cost and high starting que, there are many professional sound recorders and urate recording instruments requiring the quieter and re vibration-free performance of the capacitor synonous motor. The starting torque is less than 100 per t but the capacitor synchronous type requires no cenugal switch, making the smaller frame sizes quickly resible while driving most loads. In *Fig.* 8 is a Tagliabue lectray pyrometer powered by a miniature 1/2000-sepower synchronous capacitor motor for driving charts uniform speed, and a 1/1500-horsepower nonsynonous, reversible, dynamic braking motor operating the lancing mechanism.

Polyphase synchronous motors have even higher start- torque, with greater horsepower ratings possible in the e frames, than either of the above single-phase types d usually are preferred wherever the power supply is ilable. Their general performance is similar to the phase induction motors previously mentioned.

Synchronous motors vibrate somewhat more than nonchronous induction types. Although the rotors are dynamically balanced, this electrical vibration (sometimes called "60-cycle hum") is present to some degree depending on how saturated the particular winding may be. The starting torque also varies with the position of the rotor, therefore the rated output must be within the pull-in torque. Other typical applications are X-Ray timers, sound cameras, large clocks, and traffic signals.

Special Purpose motors, built to order for specific applications, are higher in cost, but this is more than compensated for by performance on exacting load requirements and reversing duty. One example is a torque motor designed to deliver a specified torque when stalled continuously without any danger of overheating or burning out, as would most ordinary general-purpose motors. Small torque motors with or without integral speed reducers are available in single-phase capacitor, shaded-pole, polyphase induction or special direct-current types. The locked torque and complete operating cycle, including the maximum time the motor may have to hold the load, must be known. Similarly, alternating-current and direct-current motors can be wound and connected for dynamic braking.

Features of Direct-Current Motors

Direct-Current motors are wound in three types: Shunt, compound and series. Shunt or compound motors are applicable to machines requiring constant speed, like the alternating-current split-phase or capacitor types. Shunt-wound motors have armatures with distributed windings and field coils of many turns of fine wire on salient poles. Larger ratings are usually compound-wound by adding a few series field turns, providing higher starting torque and lower inrush current at starting. Electrical connection between armature and field is made through brushes contacting the commutator.

The principal advantages of shunt or compound motors are:

1. Practically constant speed even under fluctuating loads, no racing at no load
2. Starting torques approximately 150 per cent
3. Adjustable speed, increased or decreased by suitable resistance
4. Reversible while running or at standstill.

The use of a commutator is the main disadvantage. Occasionally commutator or brush trouble will occur, especially under dynamically braked or rapidly reversed loads. These motors are used extensively on railroad and marine equipment, control apparatus, machine tools and aircraft accessories, being built for operation on various voltages from 6 to 250 volts. A typical speed-torque curve in *Fig.* 9 shows that speed is relatively constant from no load to full load.

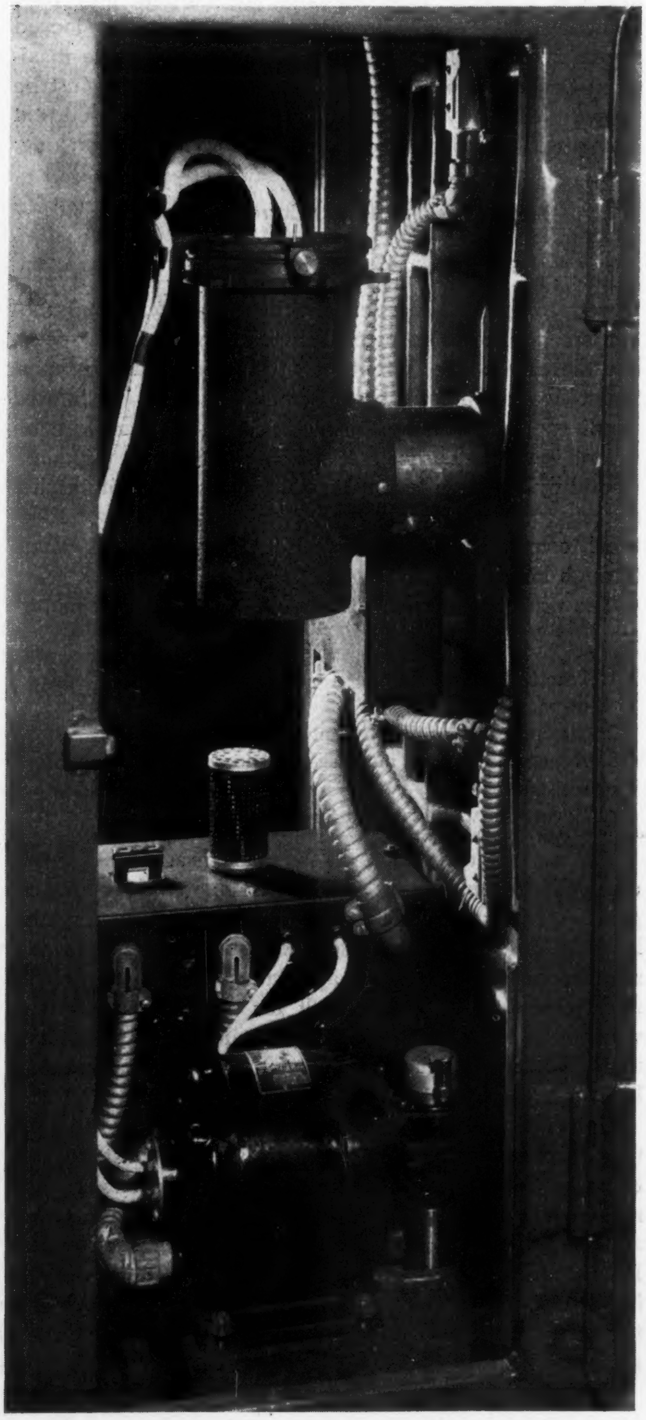
Universal series motors are popular where close speed regulation is not required, load is relatively constant, and operation is required on both alternating-current and direct-current power. Common applications include office and household appliances, and portable tools such as the Detroit surfacing sander, *Fig.* 10, direct-driven by a small, high-speed, series motor with bottom end shield machined to fit the sander mechanism and the eccentric on the driven shaft.

Series motors are wound similar to the other direct-cur-

rent types except the field is in series with the armature. Performance curves are shown in *Fig.* 11. The speed is greatly affected by even small load changes or variable voltage. Starting torque is high, and the load can be increased down to the stalling point. They are high-speed motors, developing their maximum power and efficiency at between 5000 and 8000 revolutions per minute. Principal advantages are:

1. Operation on both alternating current and direct current
2. Highest starting torque of any type motor of same rating
3. Rheostat speed control
4. High output, high speed and good efficiency combi producing maximum output for a given frame size.

Fig. 13—Worm-gear reducer motor for table drive on a contour projector is built into base of machine



When a universal motor must operate at the same on both alternating current and direct current, an ture shunt resistor may be used. Where constant within close limits must be maintained regardless of nary changes in load, voltage or frequency, an ele governor-controlled motor may be applied but mu "tailor made" to the load characteristics of the d equipment. Accurate speed control may be obtained a fairly wide range with high starting torque in the series winding for universal operation—though occa ally shunt wound for direct current only. Space doe permit more performance data; but the designer sh use governor motors only for intermittent duty, and after complete life-tests on the actual device in the

In *Fig.* 12*a* is a method to secure speed control series motor over a wide range and still retain stabili low speeds. In *Fig.* 12*b* is one way to reduce the sp of small shunt motors fairly low without sacrificing starting performance.

A split-field, 3-wire reversible, series-wound speed ducer motor with extra shunt armature lead is sh *Fig.* 13 for raising and lowering the table on a Bausc Lomb contour projector.

Series-wound motor parts are compact, dependable available at low cost for assembly in machine housi For such built-in applications, ample ventilation should provided and the motor manufacturer can supply ventilating fans. Here too, the motor engineer can be assistance and also can arrange for load tests. Series- tor parts are used on many home appliances, projec and small portable tools.

Reducer Motors Conserve Valuable Space

With space at a premium, small integral reducer mo are the answer to large positive speed reductions. Depe able and compact worm-gear reducer motors are c monly furnished in gear ratios from 6:1 up to 1120:1, ranged for mounting in practically any position. In ap ing these speed-reducer motors, care must be taken no direct couple high inertia loads to the slow shaft beca the momentum of the load during starting or stopp may damage the gears. Likewise, loads which are li to lock may strip the gears because of the tremend torque built up. If either condition is a remote po bility, some form of safety clutch or shear pin between drive shaft and the load is recommended to protect gears from damage.

To prevent excessive wear, speed-reducer motors sho be applied so that the high point on a varying load cy will not recur on the same tooth of the gear. The spe and inch-pounds torque required always should be spe fied at the slow shaft when ordering a speed-reduc motor.

High frequency miniature motors are being used on a craft and becoming more popular every year on portab tools. The good starting torque and greatly increased ou put per given frame size will result in wider applicati after the war. Also, recent developments in electron tube control of miniature motors over wide speed rang indicate unlimited future possibilities.

Stress Relief of Weldments for Machining Stability

By J. R. Stitt
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TO PREVENT distortion during machining operations, weldments used as machine parts commonly are stress relieved by heating in a furnace to a sufficiently high temperature. With the object of determining the quantitative effect of temperature on the degree of stress relief obtained, the investigation here discussed was conducted under the author's direction.

In order to produce specimens which would, because of their symmetry, build up symmetrical residual stresses, the 90-degree cross weldment shown in *Fig.* 1 was selected. Inasmuch as welds are made in the four quadrants, considerable distortion occurs when the weldment is machined, thus affording deflection values which can be conveniently measured.

After welding and stress relieving for two hours at the specified temperature, each specimen was machined parallel to the top edge to a depth of ¼-inch and the distortion measured by means of the inclined measuring fixture illustrated in *Fig.* 2. Strain gage readings at the stations indicated in *Fig.* 1 also were recorded. Additional cuts along the planes indicated in *Fig.* 1 gave a succession of distortion values from which a curve could be plotted. A series of such curves, *Fig* 3, resulted when a number of different stress-relief annealing temperatures were used. The horizontal axis of *Fig.* 3 represents distance from the top edge of the original specimen to the machined sur-

*On leave from Ohio State University. The research on which this article is based was carried out by the Ohio State University Research Foundation under an O.S.R.D. contract, for the National Defense Research committee under the direction of the War Metallurgy committee. It is reported in Ohio State University *Engineering Experiment Station Bulletin No.* 121.

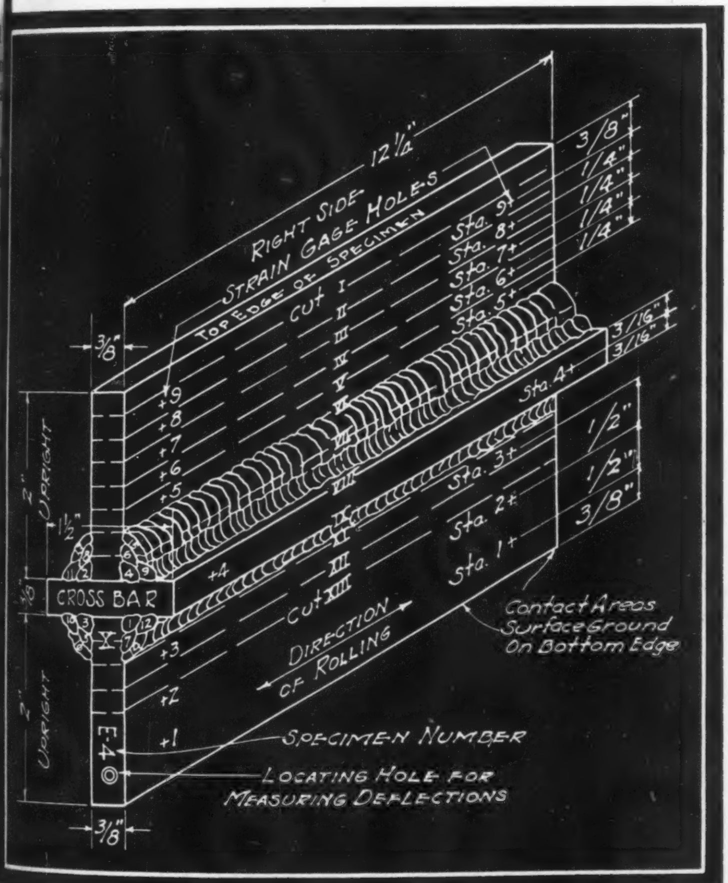


Fig. 1—Above—Typical automatically welded specimen as used in the study of the effect of stress-relief annealing temperature on machining stability of weldments

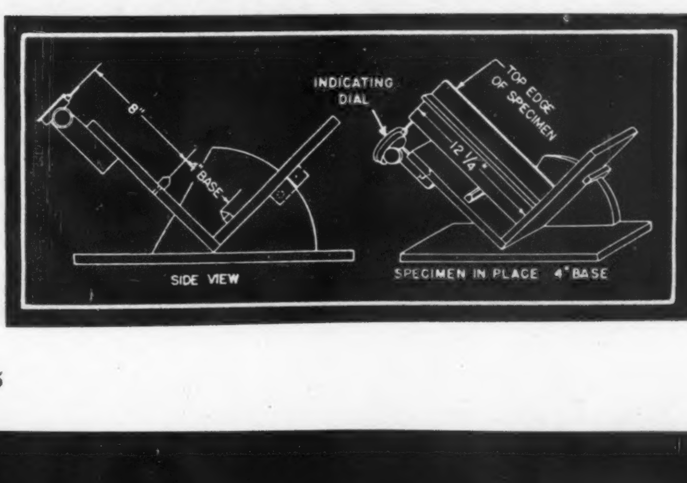


Fig. 2—Left—Inclined measuring fixture used with specimen shown in Fig. 1 facilitates accurate determination of the distortion

face, while the vertical axis is the difference in the readings of the indicating dial obtained with the measuring fixture, *Fig.* 2. These curves depict graphically the progressive improvement in machining stability obtained by stress relieving at higher temperatures.

Four different high-tensile low-alloy steels were tested in this manner; each gave a series of curves generally similar to those shown in *Fig.* 3, which is for NE 8630. Averaging the results for all four steels it was found that the relative distortion following thermal stress relief, expressed as a percentage of the highest curve ("as welded"), was as follows:

S.-R.A. Temp., F.	None	900	1000	1100	1200	1300	1400
Avg. Distortion, %	100	50	35	21	12	5	4

Some of the factors which govern the amount of distortion are: Location and sign of the stresses, magnitude of the stresses, geometry of the weldment, volume of the metal removed, location of the metal removed, rigidity of the structure, yield strength of the steel, yield strength of the weld metal, method of machining, size and location of welds, welding technique, and original tacking of the specimen.

TABLE I

Average Per Cent Reduction of Internal Stresses Due to Thermal Stress Relief Annealing

SRA Temp.	900°F	1000°F	1100°F	1200°F	1300°F	1400°F
NE-8630	53.0	68.7	78.0	87.4	95.4	95.4
SAE-4130	46.9	62.1	81.3	91.2	96.7	97.9
NAX-X-9115	46.6	64.2	76.6	88.0	93.7	95.0
NAX-X-9130	52.5	67.6	77.5	85.7	94.4	96.4

From deflection readings the magnitude of the stress relief following each treatment was calculated. A sign[illegible] cant result was the consistency of values at different [illegible] levels, leading to the conclusion that the average valu[illegible] which are shown in TABLE I, can be expected to hold [illegible] smaller cuts than those recorded in the investigation. Act[illegible] stresses remaining in the specimen after the various stre[illegible] relieving treatments are shown in TABLE II.

As a result of the investigation it is believed that we[illegible]

TABLE II

Residual Stress, psi, After Stress Relief Annealing

SRA Temp.	900°F	1000°F	1100°F	1200°	1300°F	14[illegible]
NE-8630	—21,400	—15,600	—8,000	—5,300	[illegible]000	—1,[illegible]
SAE-4130	—19,100	—14,000	—6,400	—4,800	— 100	+ [illegible]
NAX-X-9115	—19,400	—13,800	—7,200	—4,500	+ 400	+1,[illegible]
NAX-X-9130	—21,300	—15,000	—9,200	—6,100	—1,900	[illegible]

Note: —indicates relaxation of compression; +indicates relax[illegible] of tension.

ment deflection curves such as those shown in *Fig.* 3 w[illegible] enable designers and production engineers, after only [illegible] few trials, to select stress-relieving temperatures with [illegible] surance that their particular weldment will remain with[illegible] the required tolerances during and after each machini[illegible] operation, providing the weldments are machined in [illegible] manner which will not cause distortion by cold worki[illegible] of the metal.

Furthermore, if a weldment is found to distort too mu[illegible] after a certain stress-relieving treatment, say 1000 degre[illegible] Fahr., by referring to the weldment deflection curves fo[illegible] that steel it will be evident what stress relieving treatme[illegible] will be necessary on similar weldments to have the[illegible] meet any tolerance required.

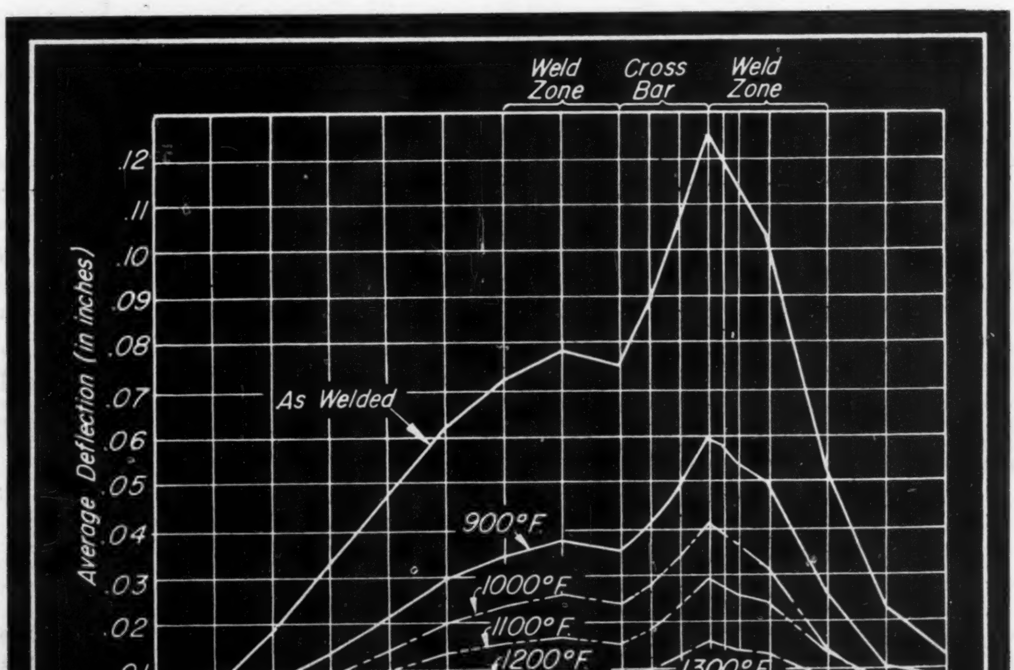
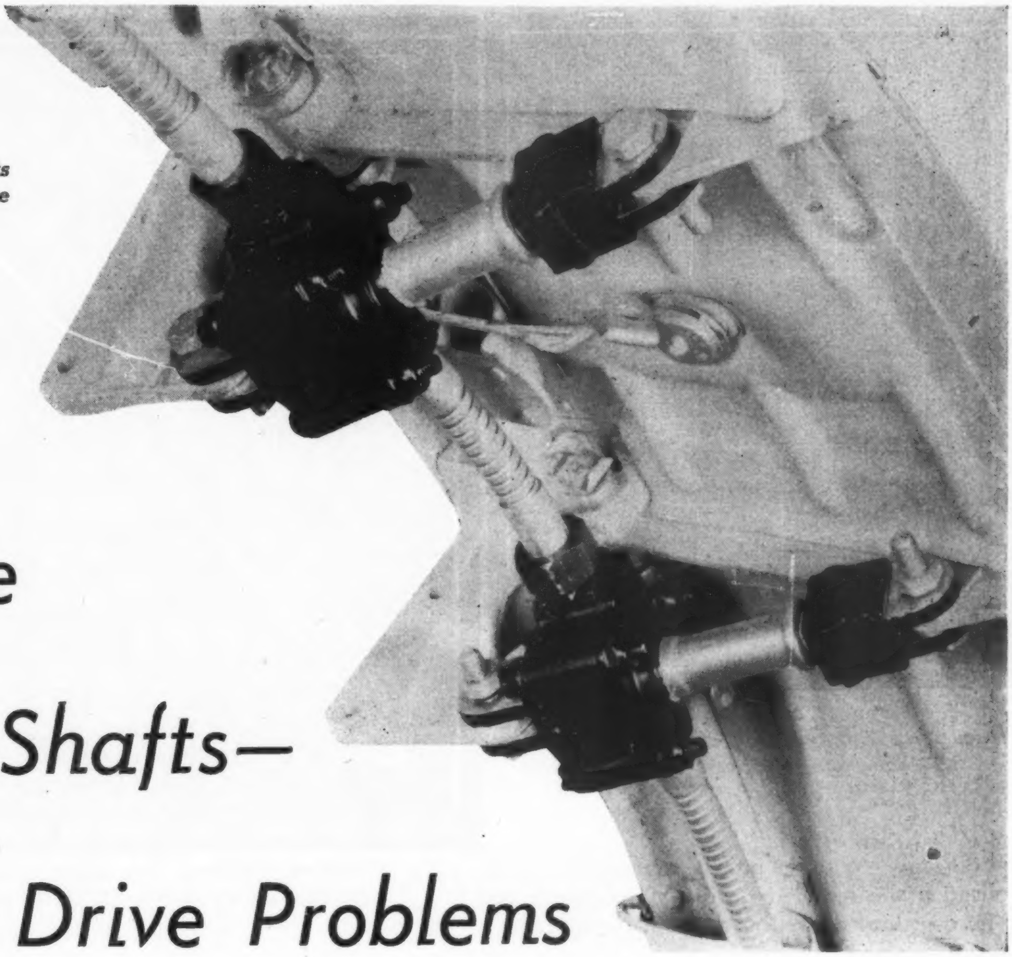


Fig. 3—Curves sho[illegible] how machining disto[illegible] tion decreases as t[illegible] stress-relief anneali[illegible] temperature increase[illegible]



g. 1—Cowl flap units
ing flexible shaft drive

lexible Power Shafts— ey to Drive Problems

By Roger W. Bolz
Assistant Editor, Machine Design

AS A MEANS for direct power transmission in machines, flexible shafts offer the designer a great many real advantages. Extreme flexibility in the lacement of driving units of this type allows improved deign, reduced installation problems, conservation of valuable space, and simplified maintenance. Flexible power hafting also fills a definite need by often making possible n a machine an operating function which otherwise would e economically out of question.

In machines which necessarily must operate under exposed and adverse conditions, the flexible shaft finds valuable application. Here the conventional bevel or worm gear drives, universal joints, belts, and like units are vulnerable. A flexible shaft, however, offers long life and comparative freedom from the ill effects of such operating conditions.

ig. 2—Flexible power shaft core construction. Central wire is wrapped with layers of spirally-wound wires

In all probability, however, the prime advantage of the flexible shaft as applied to machine power drives lies in the inherent simplicity of design. In a machine utilizing this type of drive the motor may be conveniently positioned for design purposes and easy removal or repair. The usual accuracy required in the alignment of component units is not necessary to obtain smooth, even power transmission and, regardless of obstacles or the relationship between driving and driven shafts, the flexible power connection can usually be installed with a minimum of trouble. In aircraft designing, definite influence on the ultimate performance characteristics can be obtained as a result of the concentration of heavy-weight power units and drive mechanisms at desired locations within the primary structure. *Fig.* 1 shows an open-and-close mechanism which operates engine cowl flaps of various kinds of aircraft. The flexible shaft drive allows convenient placement of the power source and also considerable latitude in the design

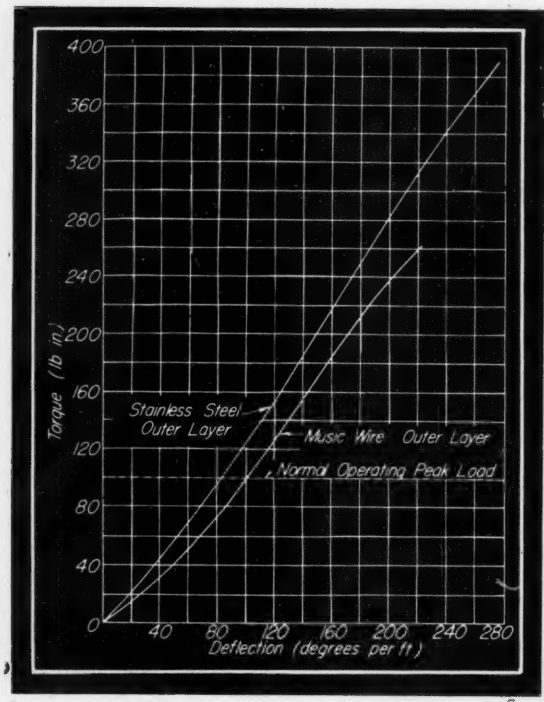


Fig. 3—Graph of static stress-strain characteristics of an 18-8, type 302, stainless-steel covered 3/8-inch core and one of music wire

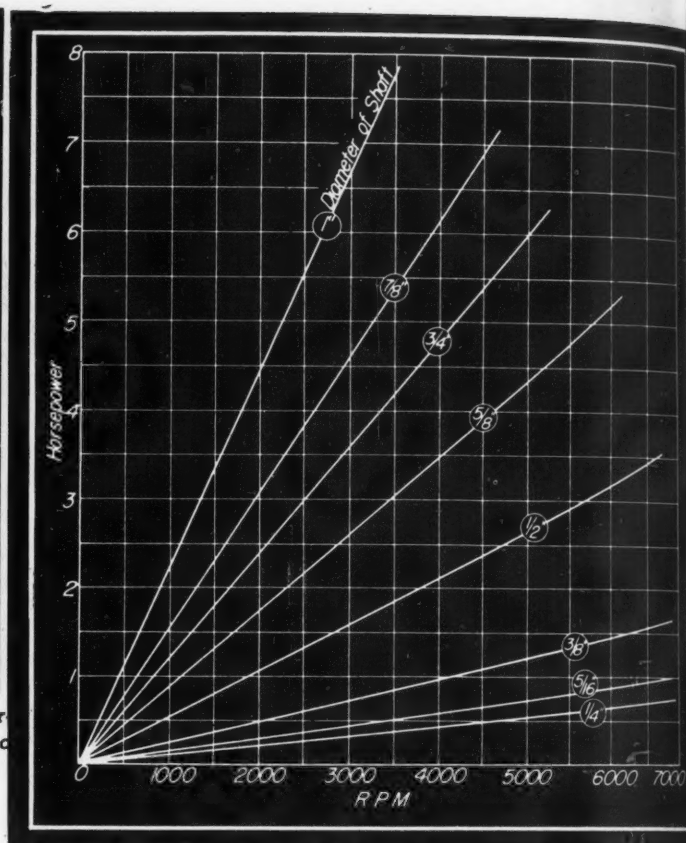


Fig. 4—Above—Nominal horsepower capacity of common sizes flexible power shafts. Core size is shown encircled

of the flaps. With the proper engineering and application the required degree of sensitivity and life, over long as well as short shaft distances, is easily obtained.

The heart of the flexible shaft is the core, or driving medium, built up from one central wire with superimposed layers of wire helically wound in alternating directions. *Fig.* 2 is a typical power drive flexible shaft core, sectioned to indicate the general construction. Power drive cores are designed for maximum efficiency in one direction of rotation only. Pitch or lay of the outer layer of wires determines the direction of rotation and should always be such that this layer tightens under the driving load. Power shaft core shown in *Fig.* 2 is for right-hand or clockwise rotation, the outer strands leading in the same manner as the threads of a left-hand screw thread. Direction of rotation for any flexible shaft installation is always determined, of course, from the driving end of the shaft. Where a drive must operate in both directions as in many aircraft installations, the shafting should be arranged so that the greater torsional load is taken in the direction of rotation giving the least deflection and greatest capacity.

Stainless Improves Strength

To adapt the cores for a wide range of loads and operating conditions, the number of wire layers, size and quantity of wires in each layer, tempering and spacing of the wires, and the winding tensions are calculated to suit. Stainless steel wire rather than regular music wire is used in the outer layer of some shafts in order to obtain the unusual advantages of this material. The stainless steel wire not only work-hardens during core manufacture, but continues to do so to some extent during operation of the shaft, creating by this action an extremely hard shell about the outer wires. Tensile strengths of these stainless wire may run as high as 350,000 pounds per square inch. *Fig.* 3 indicates the advantage of this construction in increased torque capacities per unit of deflection.

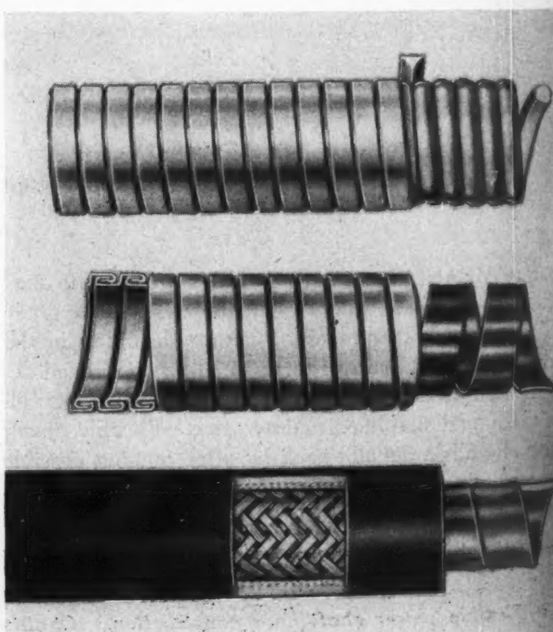


Fig. 5 — Power drive casings. Top — General purpose. Center — Heavy duty. Bottom — Heavy-duty portable to

Proper balance must be established between torque transmission capacity and transverse flexibility of a core. Maximum torque capacity is obtained by building the shaft so as to offer the greatest possible resistance to twisting while under load. Such a shaft, naturally, will have a minimum of transverse flexibility. Where a greater amount of flexibility is necessary it can be obtained, but only through some sacrifice in torque capacity. Hence, in applying a shaft the designer should use the largest radius curves possible. By placing these bends as near the input end as can be arranged, unnecessary loading of long lengths of shafting may be avoided.

An idea of the nominal operating capacities of flexible shafts up to a 1-inch size can be gathered from the chart in Fig. 4. Secondary factors such as length, degree of bend, etc., have been included for average conditions encountered. The effect of speed on the carrying capacity is clearly shown in the chart. A ¼-inch shaft at 1000 revolutions per minute has a capacity of .1 horsepower, while at 7000 revolutions per minute the capacity rises to .75 horsepower. Speeds as high as 10,000 to 12,000 revolutions per minute have been successfully used with small cores. Nevertheless, for cores of ¼-inch diameter and over it is recommended that surface speed should not exceed 500 feet per minute for maximum economy. Wherever reduction gearing is used in conjunction with a flexible shaft, it should be so placed as to allow for operation of the shaft at the highest possible speed to make available the most economical size of shaft.

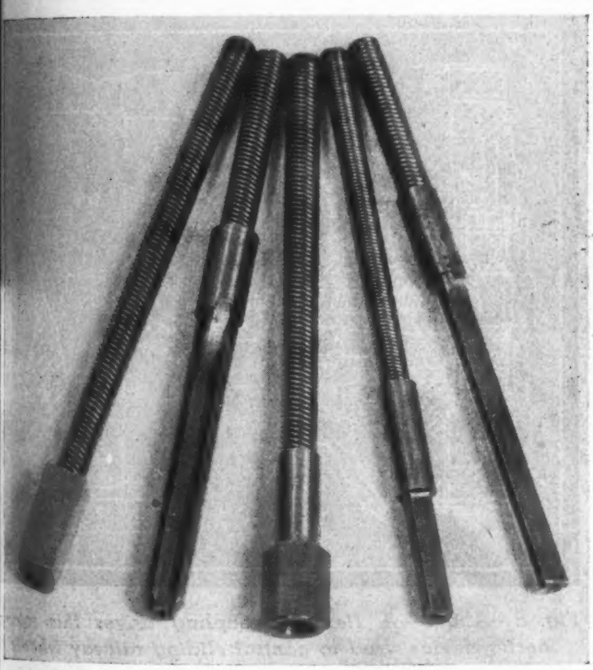
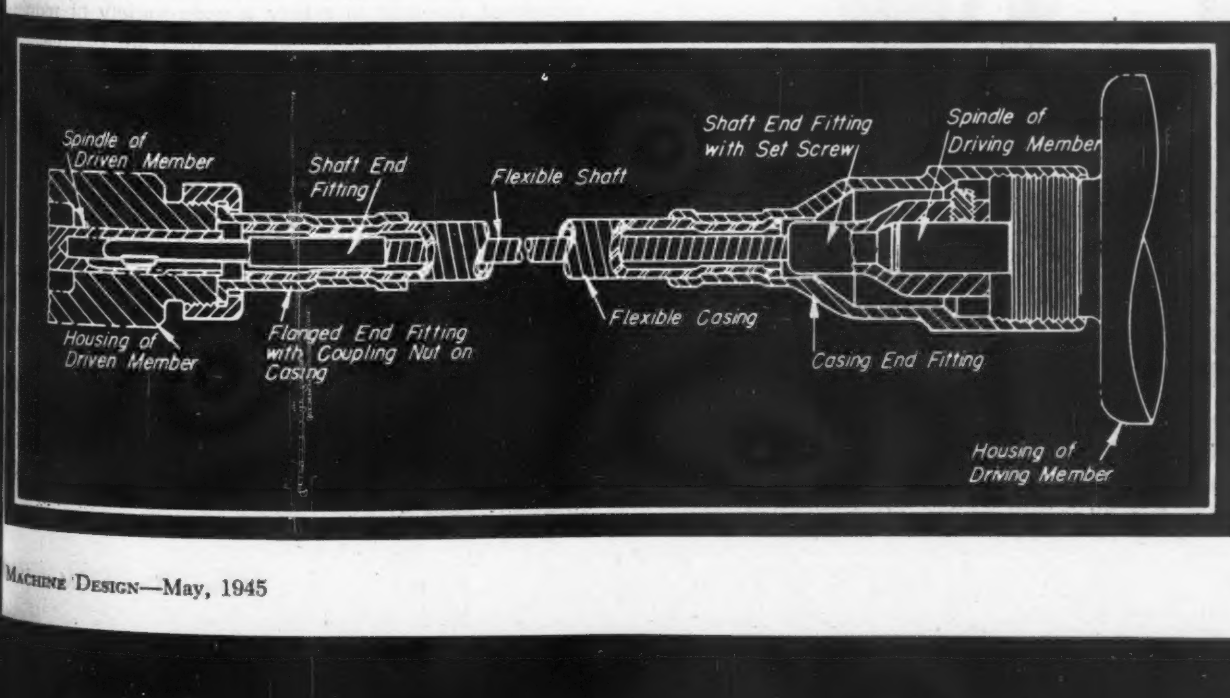


Fig. 6—Some of the more commonly used core end fittings

Except in the case of short couplings, the core is usually enclosed in a flexible casing. This casing not only acts as a bearing for long cores, preventing excessive distortion, but gives protection against hazardous conditions such as impact, abrasives, moisture, or corrosion. *Fig.* 5 shows some typical designs of power shaft casings. The general purpose type illustrated is designed for use where flexing occurs while the shaft is in operation. This casing offers an advantage in that the outside diameter is the smallest available for any given size core. Heavy-duty type casings are usually packed to retain lubricant sealed in at assembly. Wide spring-steel spiral liners used in these casings assure high efficiency under severe service requirements. The portable tool type casing offers a sturdy, abrasion-resistant rubber covering for easy handling. This casing is often used in permanent machine drives where the resilient covering is advantageous.

Some of the most commonly used end fittings are shown in *Fig.* 6, and a typical complete shaft assembly in *Fig.* 7. Standard end fittings can be used in most assemblies, but where this is not possible suitable ones can be made to fill practically any requirement. In the layout of assemblies a few important points must be adhered to in order to assure proper operation: (1) Shaft core and casing should be mounted entirely independent of each other; (2) one end of the casing should be designed for a loose nut or setscrew attachment for assembly purposes; (3) one core fitting should be rigidly mounted to the driving shaft, with the other end left free to float to absorb variations in shaft length under flexing loads; (4) core fittings should be such that the shaft can easily be inserted or withdrawn from the casing for assembly, lubrication or replacement; (5)

Fig. 7—Below—Cross section of typical shaft assembly



where fixed position drives are installed, the casing should be secured to the machine bed with suitable clamps to prevent unnecessary whip or vibration.

Shaft length in the great majority of power drive installations does not exceed 10 to 12 feet. However, successful applications have been made with shaft lengths up to 50 feet and on rare occasions some even greater. The horsepower capacities shown in the chart of *Fig.* 4 apply to shaft lengths up to and including 20 feet. In general, length does not seriously affect shaft selection until the installation exceeds 20 feet.

Frequent lubrication of flexible shafting is only necessary in the larger sizes. However, some lubrication at regular intervals is desirable for all shafts, depending upon the type of service and core construction. Tests show that stainless steel cores generate less friction and heat in operation and consequently lubrication problems are somewhat reduced. These cores are also highly resistant to galling, actual use indicating little or none at all even when lubrication fails. Dissimilar metals combined with extreme hardness appear to reduce normal shaft wear to a minimum.

Provision for lubrication of a flexible shaft should be made to allow the entire unit to be disassembled and the core and casing thoroughly cleaned in some suitable solvent. As the core is being replaced in the casing it should be covered with a neutral vaseline or a good grade of medium nonfluid oil. No graphited oil or grease should be used.

Flexible shaft couplings provide the qualities necessary in good coupling practice with the additional advantage of being able to absorb extreme amounts of misalignment. A particularly interesting drive utilizing these advantages is the Westinghouse "Declostat" shown in *Fig.* 8. This unit is employed to control the brakes of railway passenger cars. Directly connected to the axle member by means of a flexible shaft coupling, this rotary inertia device acts to control braking deceleration and prevent slipping or sliding of the wheels. Providing a continuous direct drive, the coupling nevertheless permits the usual end play required in the axle assembly and is unaffected by any misalignment or journal wear that may develop.

The special drilling machine shown in *Fig.* 9 embodies a versatility and adaptability not usually found in ordinary drilling machines. Any or all of the 38 flexible shaft spindles may be used simultaneously in groupings and dimensional spacings to satisfy a wide variety of requirements. Extremely close hole spacings are possible. In the illustration thirty-two No. 32 drill size holes are being made at one time in an intricate gas burner casting.

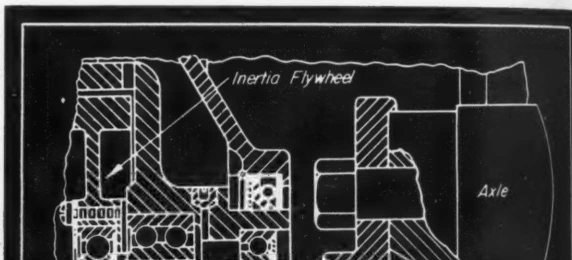


Fig. 8—Above—A flexible coupling drives this rotary inertia device used to control sliding railway wheels

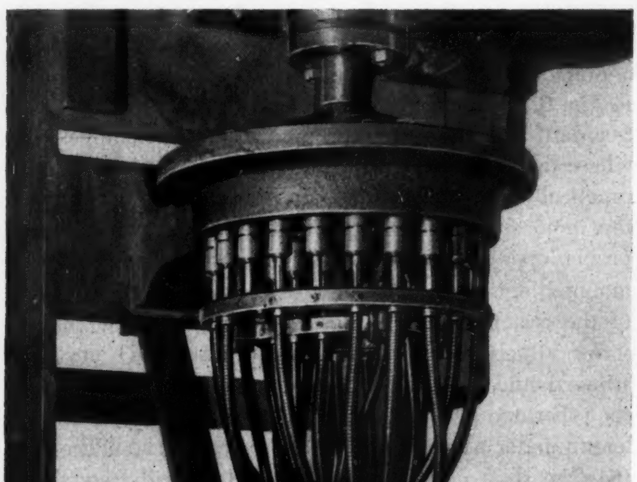


Fig. 9—Flexible shafts provide unusual versatility in this 38-spindle universal drilling machine

It can be seen from the foregoing that whenever new or improved design is under consideration, the flexible shaft merits attention as a means of direct power transmission. It may easily provide the final key to the achievement of a successful design where space limitations, operating conditions, relative movements, or positions of parts present a major handicap.

Collaboration of the following companies in the preparation of this article is acknowledged with much appreciation: Elliott Mfg. Co. (*Fig.* 3); General Gas Light Co. (*Fig.* 9); Lear, Inc. (*Fig.* 1); Mall Tool Co.; F. W. Stewart Mfg. Co.; Stow Mfg. Co. (*Figs.* 2, 4, 5 and 6); S. S. White Dental Mfg. Co. (*Fig.* 7); Westinghouse Air Brake Co. (*Fig.* 8).

When Few Parts Need Balancing

By Lawrence E. Steimen

Engineering Consultant, Research Div. United Shoe Machinery Corp.

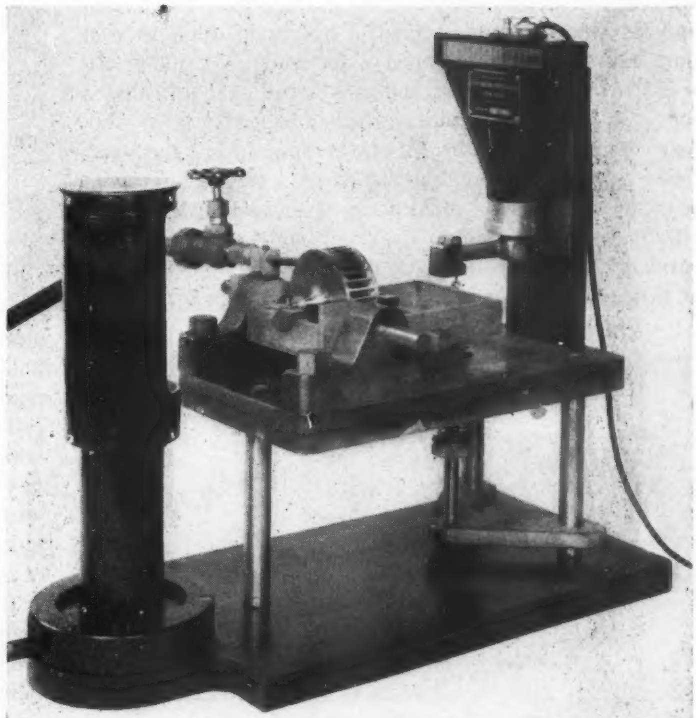


Fig. 1—View of simple dynamic balancing apparatus shows rotor in position for balancing. Adjustable spring permits centering the amplitude-indicating light beam on the scale

ALTHOUGH any rotor may be dynamically balanced by the addition or subtraction of weights in each of two selected end planes, the question is how to find both the position where the correction is needed and the amount of correction necessary*.

Recently the author was faced with the problem of attaining fine balance in a limited quantity of small rotors designed to run at speeds in excess of 25,000 revolutions per minute, without a commercial balancing machine conveniently available. After some investigation it was discovered that extremely accurate dynamic balance could be obtained in a reasonably short time by the use of certain laboratory apparatus on hand and without any intricate or expensive construction. The following account of the method, apparatus, and results is offered as being of possible assistance to others who may have occasional precision balancing problems in insufficient volume to justify a commercial machine.

How Rotor Is Mounted for Test

Method is to mount the rotor (without its ball bearings) in plain bronze half bearing blocks fixed in a light wooden frame, *Fig.* 1. The frame is supported on frictionless steel pivots so that the fulcrum line of action is coincident with the rear end of the rotor, the front end of the frame resting on a fully adjustable light spring. This permits only the unbalance near the front end plane of the rotor to induce a disturbing force in the spring through the wooden frame. A vibroscope is used to indicate the amplitude of this vibration. The vibroscope pin rests on the wooden frame and, in moving with the frame, rotates a small mirror which focuses a light beam on a ground-glass screen through a lens system giving approximately ½-inch visual movement of the light beam for a one-thousandth inch deflection of the frame.

The vibroscope was found to be the simplest means of recording the amplitude after both a magnetic and crystal pickup were tried. A capacitance type of pickup was contemplated as possibly being more sensitive than the vibrometer, but the vibrometer proved to be satisfactory for the job at hand and no further work has been done on electrical pickups, although any one of the several types might be used.

The rotor, which is designed to be air-driven in actual use, is also driven by air in the test frame. A reducing valve provides speed control, with an ordinary hand-operated valve inserted in the line after the reducing valve to provide vernier speed control.

A stroboscope lamp is used to illuminate the rotor and is connected to flash from the 110-volt, 60-cycle line. A single indicating spot of Prussian blue is marked on the rear end plane of the rotor to be tested and the rotor is then brought up to 3600 revolutions per minute, at which speed the indicating spot is "stopped". (On rotors that are balanced at 1800 revolutions per minute, the single indicating spot appears as two spots "stopped" 180 degrees apart). The speed selected for balancing changes neither the amount nor the position of unbalance, but it must be

*W. I. Senger—"Specifying Dynamic Balance", MACHINE DESIGN, Nov., 1944.

remembered that when running below the critical speed the high spot or extreme limit of amplitude will be the "heavy" spot, and conversely when running above the critical speed the high spot is the "light" spot. Whatever speed is selected for balancing should be used during the entire balancing procedure on that particular rotor, as all comparisons of amplitude must be made at the same speed.

In order to determine the location of unbalance, some arbitrary correction weight is put in the plane of balancing and several runs made with the weight in different positions along the circumference of the rotor. Modeling clay was used for the weight and was found to hold satisfactorily on an oil-free rotor surface at speeds up to 3600 revolutions per minute. A curve representing the variation in amplitude of vibrations, as read on the vibrometer scale, with the angle of location of the weight can be so obtained. The minimum amplitude indicates the true location of the "light" spot. With the modeling clay placed at this location, more clay is added or some is removed as necessary until the minimum amplitude has been reduced as close to zero as possible.

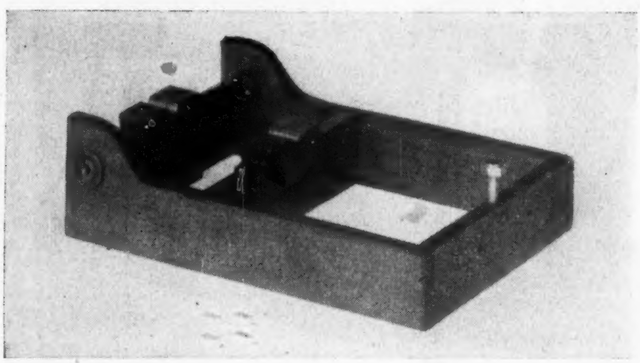


Fig. 2—Close-up of wooden frame in which rotor is mounted shows relative location of bearings and fulcrum

To make the correction in the rotor, the amount of unbalance is computed from the combined weight of the modeling clay and its radial distance from the axis of rotation, and a hole drilled 180 degrees from the scribed location. Size and depth of the hole are such as to remove a weight of metal at a radial distance to equal the value of the unbalance.

After making this correction the rotor is reversed in the frame so that the rear end plane now becomes the front end plane, and the procedure is repeated.

After making this second correction in the second plane, the rotor is again reversed and run to check it in the original plane, as sometimes a very slight recorrection is necessary.

One typical rotor gave a scale reading of 11 units when run at 1800 revolutions per minute as received with no correction added. When an arbitrary correcting weight of modeling clay was placed on the end plane circumference and positions plotted against amplitude, the minimum amplitude was found to be 5 scale units. Additional clay was added to the modeling clay in this location to reduce the scale reading to ¼-unit and the angular location was scribed on the rotor. After drilling the proper size hole at 180 degrees to this location, the rotor was then reversed in the frame and the procedure repeated on the other end plane, after which the first plane was again rechecked. Results of the balancing operation are shown in the following table:

	Unbalance (in.-oz)	Cent. Force at 25,000 rpm (lbs)
Rotor as received	.03024	33.5
Rotor after balancing	.0007	.776

Remaining unbalance is computed by comparing .03024 inch-ounces, which caused a deflection of 11 units, with the unbalance corresponding to ¼-unit remaining after the balancing operation, thus .03024 × ¼/11 = 0.0007 inch-ounce.

Centrifugal force due to unbalance is given by the following relation:

$$CF = 1.774 \times 10^{-6} \times wr \times N^2$$

where wr = unbalance, inch-ounces and N = speed, revolutions per minute. Hence the centrifugal force after balancing is $1.774 \times 10^{-6} \times .0007 \times 25{,}000^2 = .776$ pounds.

To accommodate rotors of different sizes and shapes, a number of the simple wooden frames may be built, *Fig. 2*. Better still, a light-weight aluminum frame may be built with interchangeable bearing blocks and adjustably slidable bearing block supports to accommodate a large range of rotor sizes.

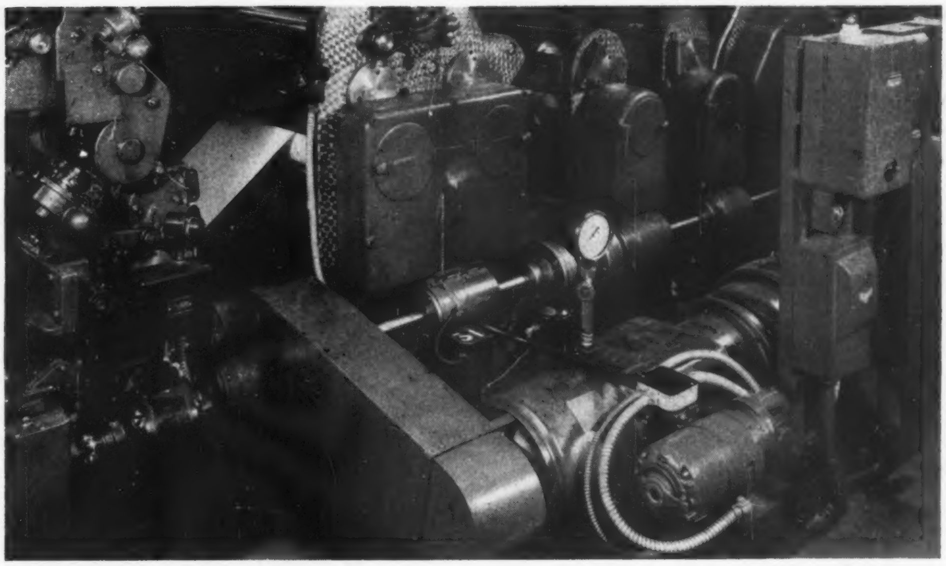
It was found that during the balancing process the exposed shaft ends of the rotor, which are of soft metal, had a tendency to lap undersize even though the bearings were lubricated. To correct this conditon, the rotors were designed with extra long shaft ends and a magnesium bushing, shown in *Fig. 2*, was used as a filler piece to protect the portion of the shaft which ultimately carries the ball bearing. After balancing is completed, the shaft extensions are cut off.

Complete balancing of this rotor with the results indicated required two hours, but it has been found that less time is required as the operator gains more skill and confidence. The ultimate accuracy of this method, in common with all balancing procedures, depends to a considerable degree upon the operator and could perhaps be further refined if a closer approach to perfection is required.

DURING the four and one-half years from July 1, 1940 to December 31, 1944, U. S. production of selected items of war equipment was as follows:

Item	Number	Weight
Heavy Bombers	28,471	607,899,000 pounds airframe
Fighter Planes	79,776	412,589,000 pounds airframe
Transport Planes	19,547	192,356,000 pounds airframe
Naval combat Vessels	1,091	2,985,000 tons displacement
Landing Vessels	51,364	2,543,000 tons displacement
Maritime Vessels	4,631	45,384,000 tons deadweight

Also manufactured were 75,204 tanks, 14,767 armored cars, 110,945 trucks over 2½ tons and 658,523 trucks under 2½ tons. Communication and electronic equipment was produced to the value of $9,405,000,000.



g. 8—Hydrostatic variable-speed drive in foreground incorporates variable-delivery pump and constant - displacement motor in one housing. It used to drive a special continuous-forms printing press. — Photo, courtesy The Oilgear Co.

Selecting Drives for Speed Control

By E. L. Schwarz-Kast
Armour Research Foundation

Part II—Hydraulic

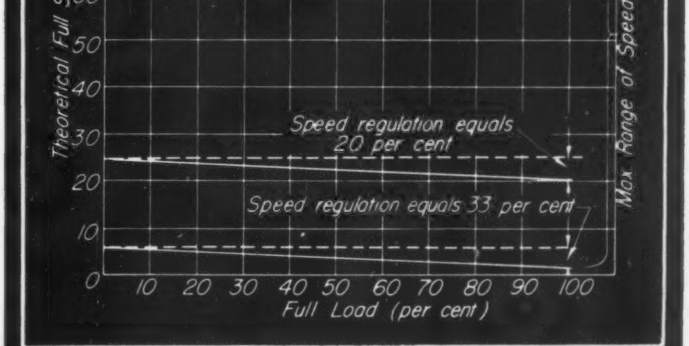
THERE are two main systems of hydraulic variable-speed devices, the hydrostatic and the hydrokinetic types. First to be developed was the hydrostatic transmission, where a liquid is put under pressure by a motor-driven pump and passed through a hydraulic motor. Power is transmitted by the fluid as a result of the pump delivery and the hydraulic pressure. Assuming the efficiency of the hydraulic motor to be 75 per cent, the horsepower output of the hydraulic motor is

$$HP_{output}=\frac{.75pq}{1714}=\frac{pq}{2285}$$

where p is the hydraulic pressure in pounds per square inch and q is the fluid flow in gallons per minute. The fluid also carries kinetic energy, but the amount is small because of the low velocity and the small quantity of liquid involved. Also there is no change in the kinetic energy content inasmuch as the liquid leaves the motor with the same velocity as that with which it entered; consequently in this system the kinetic energy of the liquid is not used at all.

Somewhat later the second group, the hydrokinetic type, was developed. This system consists of a radially-vaned impeller connected to the driver and

Fig. 9—Approximate indication of range of speed variation and magnitude of regulation of a hydrostatic variable-speed device



a similarly radially-vaned runner on the driven end. Impeller and runner face each other without mechanical connection. The liquid leaves the vanes of the runner at a much lower velocity than when it entered and the power is transmitted to the runner by the kinetic energy of the liquid.

A thorough discussion of both these groups would exceed by far the scope of this article, but a brief review of their main characteristics, being of general interest, will follow.

Features of Hydrostatic Systems

Of the various hydrostatic variable-speed systems available, all have the same principle and employ one of the following combinations:

a. A constant-pressure, constant-speed, variable-delivery pump and a constant displacement hydraulic motor.

b. A constant-delivery pump and a variable-displacement motor

c. A variable-delivery pump and a variable-displacement motor.

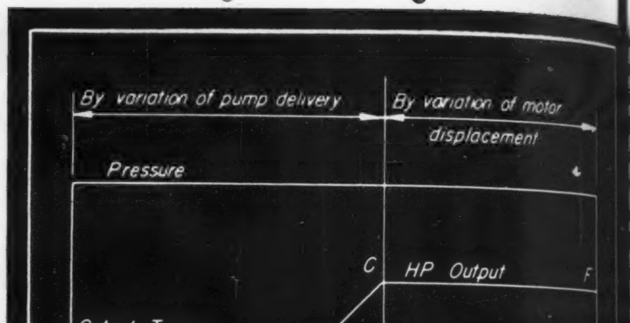


Fig. 10—Pressure, output torque and horsepower, and pump delivery versus speed for hydrostatic variable-speed devices

The idea of using hydrostatic variable-speed drives appeared in this country probably for the first time in 1905 in U. S. Patent No. 797216 covering the hydraulic drive of a reciprocating carriage travel for a grinding machine (5)*. Then during a long period of years it was entirely forgotten. About 1927 hydraulic drives reappeared in the design of machine tools (6). Since then application has been increasing steadily, *Fig.* 8. It is taken into consideration whenever a wide-range gradual-speed variation is required for either linear reciprocating applications or for rotating drives.

Success of hydrostatic variable-speed drives is well founded on the following important advantages.

1. Efficient power transmission without complex mechanisms from a constant-speed prime mover to a reciprocating or rotating movement
2. Flexibility in locating the driving and driven elements of the drive
3. Adaptability to remote control
4. Gradual and stepless speed variation in a wide range from maximum speed down to near zero speed in either direction, with excellent matching of speeds to the requirement
5. Smooth, stepless, uniform and quick acceleration from zero to maximum speed
6. Quick, but cushioned reversing, even with large masses
7. Inherent overload protection—drive can be stalled indefinitely under load without damage
8. Smooth operation, without vibrations or shocks; load peaks and vibrations damped and not transmitted to the power supply
9. Unit shows endurance and only slight wear, due to simplicity and few moving parts which are lubricated by the hydraulic fluid.

The only disadvantages are that a certain degree of care is required to keep pipes, fittings and seals tight. Proper oil of adequate viscosity must be provided. In cases where changes in ambient temperatures are great, the change in viscosity of the fluid might sometimes be inconvenient. First cost is high, but is about the same as that for electric variable-voltage control which, in many respects, has similar operating properties.

Before discussing speed-torque characteristics at different adjusted speeds (speed regulation), the significance of slip and the factors which affect it will be treated.

In all hydraulic drives there is a difference between theoretical and actual speeds. This difference, usually expressed as percentage of the theoretical speed, is called slip and is caused first by leakages and second by incomplete filling of the pump cylinder during the suction period.

Losses Vary With Speed

Leakage losses occur at piston rings, stuffing boxes and the like and can never be completely avoided, because it is natural that moving parts have a certain clearance. This loss increases with increasing pressure and decreases with increasing speed.

Filling losses increase with increasing pump speed and can be kept small by using a low-speed pump, by using the proper viscosity oil and by keeping the amount of entrained air as low as possible. This can be done by arranging the pump below the oil level in the tank (the level in the tank should be the highest spot in the hydraulic system), by a careful seal of the suction pipe line, by extending the oil return line permanently below the oil level in the tank, and by avoiding air pockets in the suction line and sudden changes in direction or profile of the flow.

Although it is known that the slip as a whole increases with decreasing speed and also increases with increasing load, not much has been published concerning its actual magnitude and relation to speed and torque. In many

*Numbers refer to references given at end of article.

es the presence and magnitude of slip or regulation is t of importance, but there are other cases, especially th machine tools and testing machines, where the knowl- ge of accurate speed-torque characteristics is of great erest or even of decisive importance. For all electrical stems, as will be discussed in the next article in this ries, thorough information on this point is available. is might be a reason for preferring electrical systems in ch cases. It seems that so far the manufacturers of hy- aulic devices have not yet realized the importance of is subject and that more information about speed regu- tion would doubtless facilitate the use of the hydrostatic stems in these critical cases.

In the diagram, *Fig.* 9, is given an approximate indica- n of the range of speed variation and the magnitude of gulation obtainable by a hydrostatic device.

As mentioned, the speed of a hydrostatic system can be ried by one or both of two means: Changing the pump livery and changing the displacement of the hydraulic otor. The diagram in *Fig.* 10 shows the relations be- een pressure, output torque, output horsepower and mp delivery during both these phases of speed control. he following characteristics of the drive will be evident examination of the diagram.

Methods of Speed Variation

When speed is changed by variation of the pump de- very (usually by changing the piston stroke), the output rque remains constant over the whole range and the out- ut horsepower varies in proportion to the speed. By in- easing the pump stroke to its maximum the pump de- very rises to point *A*, the speed to I and the output horse- ower to *C*.

Speed increase beyond point I may be obtained by re- ction of the displacement of the hydraulic motor (by educing its stroke). During this phase the output torque ecreases with increasing speed and the output horse- power remains practically constant. The final maximum speed is II, the output torque *E* and the horsepower out- put *F*.

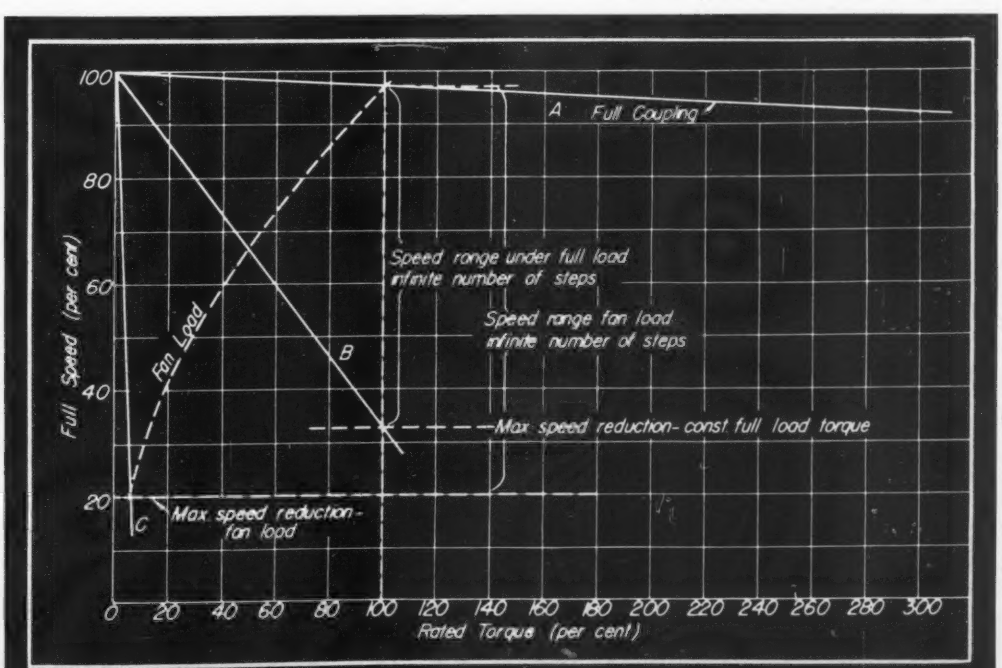
Some years back the electrical variable-voltage control was commonly recognized as the outstanding effective method for gradual, stepless, wide-range speed control, without any extra wear or heat to be dissipated. During the past ten years the hydrostatic methods, having similar characteristics and beyond these some favorable features not offered by the electrical methods, have grown to become serious competitors. The present fields of application, only to mention a few, include drives for machine tools, presses, printing presses (Fig. 8) and steering mechanisms on ships. There are also excellent prospects for further developments in the future.

Characteristics of Hydrokinetic Units

Hydrokinetic variable-speed devices were discussed in a recent article (7), but for the sake of completeness the salient features and performance characteristics of such drives will be discussed briefly. Hydrokinetic drives are of two types which will be referred to in the following as "couplings" and "transmissions", respectively. In the coupling the output speed and ratio between output speed and input speed can be varied by changing the amount of fluid in the circuit, but the unit is incapable of torque conversion, the output and input torques being the same. Hydrokinetic transmissions, on the other hand, are capable of effecting torque conversion between input and output but, with units at present available, the speed ratio is beyond the control of the operator. Speed ratio automatically adjusts itself according to the output load and input power.

In the variable-speed coupling, changes in the amount of fluid in the working circuit are effected in one of two ways. In the first a small reversible rotary pump arranged in the fluid circuit supplies oil to or removes oil from the

ig. 11—*Right*—*Speed- rque characteristics f a variable-speed hy- drokinetic coupling*



coupling. In the second type the quantity of fluid in the working circuit is changed by an adjustable scoop tube, no auxiliary pump being necessary.

For both types of coupling the speed-torque characteristics are identical and are shown in the diagram, *Fig.* 11. Curve *A* gives the speed versus torque when the coupling is full. Curve *B* shows the maximum speed reduction under full load, while between *A* and *B* an infinite number of steps is available. Further reduction below *B* under full load is not recommended for the following reason: With this device the total slip energy is transformed into heat which has to be dissipated by the circulating oil. This heat dissipation is the limiting factor and is the reason why at full torque no greater reduction than 67 per cent can be handled, corresponding to the quantity of oil being circulated and available for cooling purposes. With a "fan" load where the torque varies as the square of speed, a further reduction is possible down to curve *C*, about 80 per cent reduction. Between speeds *A* and *C* there is also over the whole range an infinite number of steps available.

Advantages of Couplings and Transmissions

As shown in *Fig.* 11 the full speed *A* is practically constant, independent of the load. Slip, the difference between no-load and full-load speed, is about 3 per cent at rated load. The adjusted reduced speed remains constant only under a constant load because when the load changes the adjusted speed changes too. The drive is analogous to the electrical method of speed reduction by series resistances.

Important advantages of these hydraulic couplings are:

1. Gradual stepless speed control over a fairly wide range
2. Perfect clutch action
3. Overload protection
4. Isolation of load peaks, shocks or vibration from the prime mover
5. Suitability for use with a squirrel-cage or synchronous motor having a low starting or pull-in torque, when used with heavy starting loads.

Hydrokinetic transmissions, capable of effecting conversion of torque between input and output shafts, employ a stationary element to absorb the difference in torque. Driving element or impeller usually is single stage while the driven element or turbine may have one or m... stages. Typical speed-torque characteristics for such unit in association with a power source having appr... mately constant speed, such as a squirrel-cage motor, shown in the diagram, *Fig.* 12, and are representative a unit with three stages in the turbine. At maximum ... put speed (or runaway speed) the ratio of output to ... put speed is about .8, but under this condition the out... torque and efficiency would be zero. Over the useful ... erating range indicated in *Fig.* 12 the efficiency exce... 75 per cent. Continuous operation below and above ... range involves power losses which call for special cool... arrangements. At starting a relatively high output torq... is available due to the torque conversion feature, wi... out imposing an exceptional load on the motor.

Outstanding advantages of the hydrokinetic transm... sion include the following:

1. Isolation of load peaks, shocks or vibration between driving and driven units
2. Stepless speed change, having automatic speed reduction under increasing load and speed increase under decreasing load, without any mechanical adjustment, equivalent to the speed-torque characteristics of a series motor
3. Ability to hold stalled or inching loads with torque increase.

Opposing these advantages is the disadvantage th... speed *ratio* is not under the control of the operator. Ho... ever, with a variable-speed prime mover, such as an i... ternal-combustion engine, the actual *output speed* can ... course be varied at will by controlling the speed of ... prime mover, while still retaining the torque conversi... feature. By providing adjustable stationary blades ... speed ratio can be varied independently of the torq... should the advantage justify the attendant mechani... complication.

Electrical variable-speed units will be the subject ... Part III of this series, to be published in the next issue.

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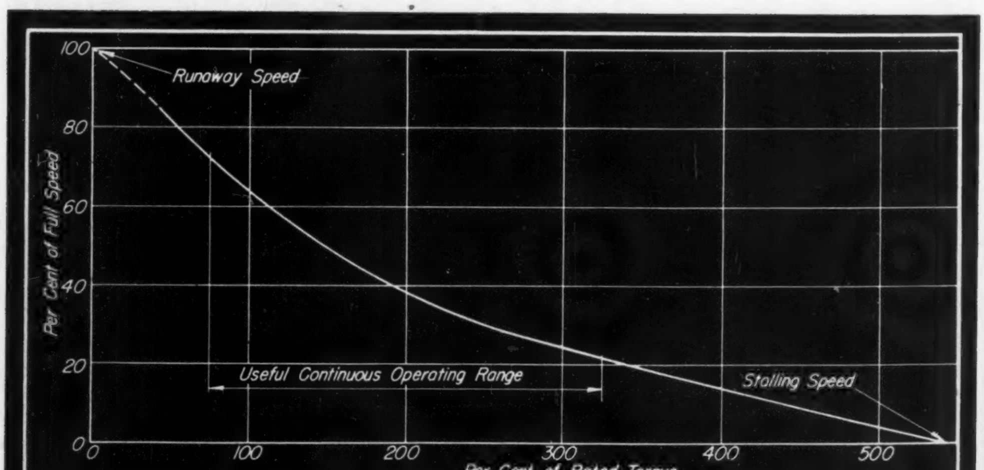


Fig. 12—Left—Spe... torque characteris... of hydrokinetic tra... mission or torque c... verter, showing use... continuous operati... range

Predicting Power Losses in Journal Bearings

By Charles D. Wilson
Steam Turbine Department
Allis-Chalmers Mfg. Co.

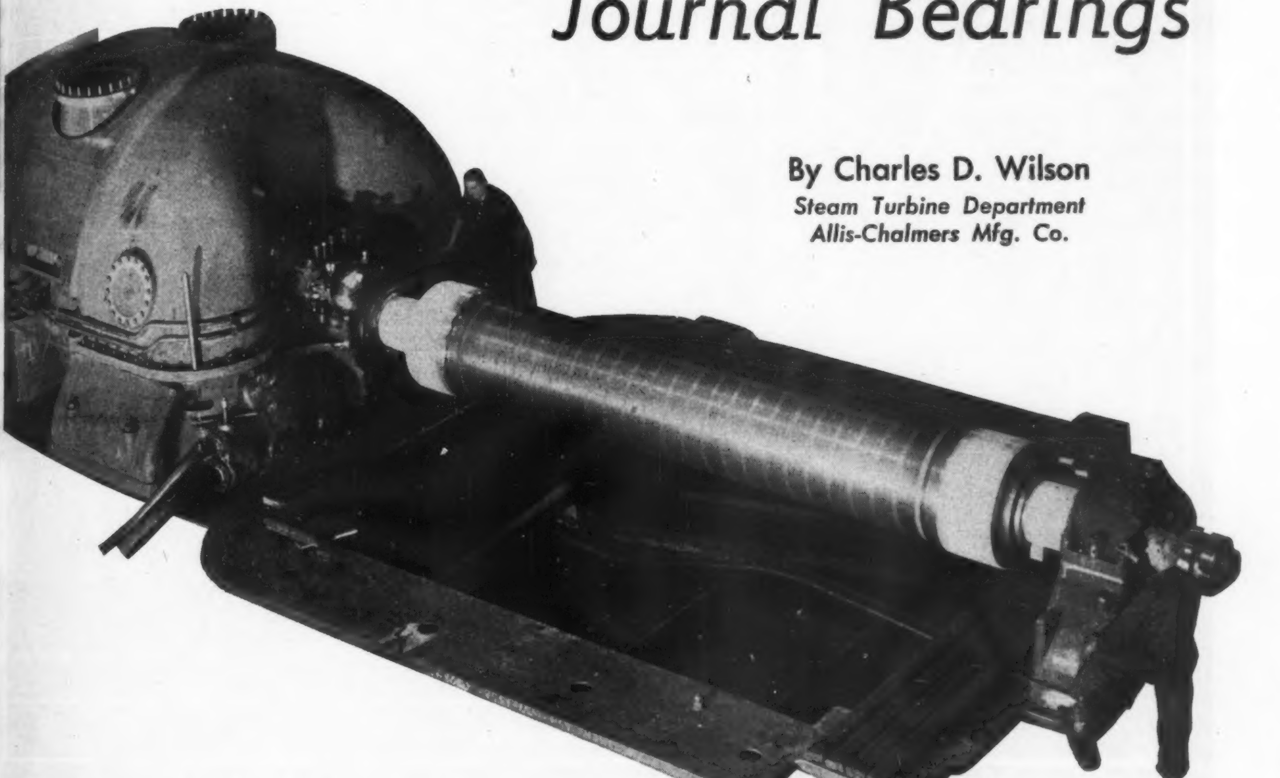


Fig. 1—Test setup shows 35,000-kilowatt steam turbine unit running at 3600 revolutions per minute in 13½-inch diameter bearings with a surface speed over 144 mph

POWER losses in large bearings are greatly influenced by the rate of oil circulation and by the design of the unloaded portion of the bearing. This has been indicated by extensive testing of full-size bearings in 1800 and 3600 revolutions-per-minute machines under actual operating conditions. Such problems as estimating power loss in bearings and determining the oil flow required to get a desired temperature rise in the oil passing through the bearings now are being worked out using curves derived from calculation and interpretation of test data, as discussed in the present article.

Diameter of large turbine bearings operating at speeds of 1800 and 3600 revolutions per minute usually is governed by the size of shaft necessary to transmit the torque, due consideration also being given to allowable bending stresses and shaft deflections. Present practice for this type of service limits the maximum unit load on the projected area of the bearing to about 200 pounds per square inch. As a result, the ratio of bearing length to bearing diameter on modern turbines frequently works out to be about unity with the lower limit around .8 and the upper limit approximately 1.3.

On most direct-drive turbines the bearings are designed to operate with turbine oil having a viscosity of 150 seconds Saybolt Universal at 100 degrees Fahr. Diametral clearance between the bore of the bearing and the turn on the journal is usually between .001-inch and .002-inch per inch of bearing diameter.

For the foregoing conditions the problem of maintaining an adequate oil film usually is not a difficult one. Speed retardation curves taken on large rotors coasting down from full speed indicate that an oil film is maintained until quite low speeds are reached. *Fig.* 2 is a typical curve of this type for a 3600 revolutions-per-minute rotor weighing 52,000 pounds and running in 13½-inch diameter bearings. When the data were taken, the only forces retarding the rotor were the rotor windage and the friction in the bearings. The change in slope of the curve occurring at about 40 revolutions per minute indicates the point where the oil film started to break down. Actual measurements of oil-film thickness on a 9-inch diameter × 10½-inch long bearing, shown plotted as a function of speed in *Fig.* 3, indicate that after the oil film was established at a comparatively low speed it increased in thickness at a uniform rate with increase in speed.

In order to run large high-speed bearings without overheating, it is necessary to use an oil cooler and forced circulation, and to cool the bearings by circulating considerably more oil than the minimum flow required for lubrica-

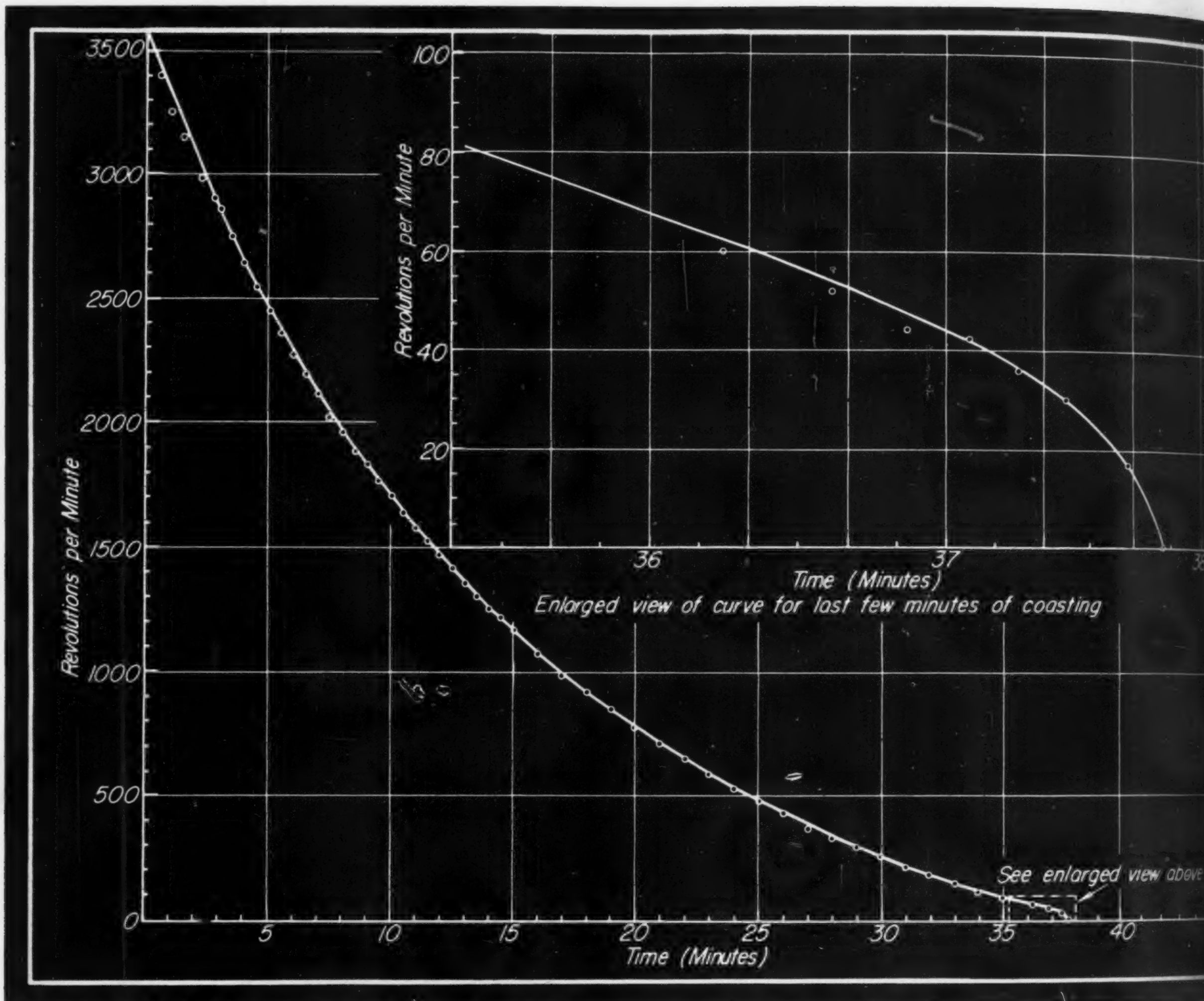
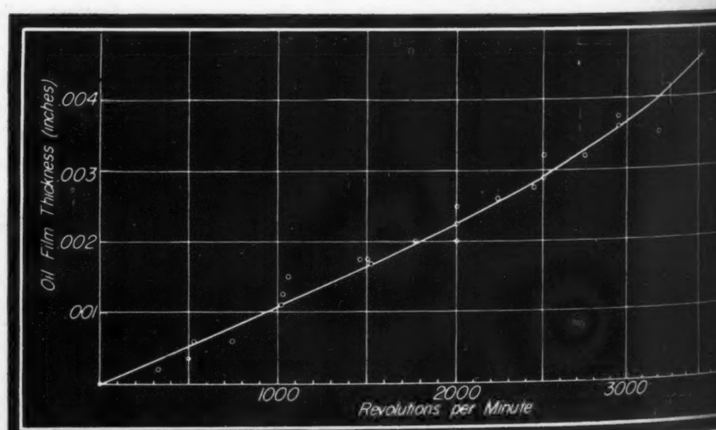


Fig. 2—Above—Speed retardation curve for unattached 52,000-pound rotor subject to rotor windage and bearing friction

Fig. 3—Below—Oil film thickness as a function of revolutions per minute for 9-inch bearing carrying 12,750 pounds total load



tion. Present practice for turbine bearings usually limits the allowable temperature rise of the oil passing through the bearings to between 20 and 30 degrees Fahr., with a normal oil supply temperature of between 110 and 120 degrees Fahr. While most turbine bearings will operate satisfactorily with temperatures considerably higher than these, the lower temperatures are used because they give a greater factor of safety and also tend to reduce the rate of oil oxidation and lengthen the life of the oil.

One problem in designing large turbine bearings is estimating the rate of oil circulation required to maintain a desired temperature rise in the bearing. Experimental data to assist in making these estimates were compiled from a series of tests in which the oil flow to the bearings was measured with positive-displacement meters and the average temperature of the oil entering and leaving the bearings was checked. Oil supply temperatures were controlled by regulating the flow of hot and cold water to a heat exchanger in the oil supply line, and oil flows were varied by adjusting valves at the inlets to the bearings.

In these tests the bearing loads observed were limited to the range covered by standard practice, but the speeds were varied over the whole range up to maximum operating speed, and data were taken with oil temperatures and oil flows considerably above and below normal. No operating difficulties were encountered.

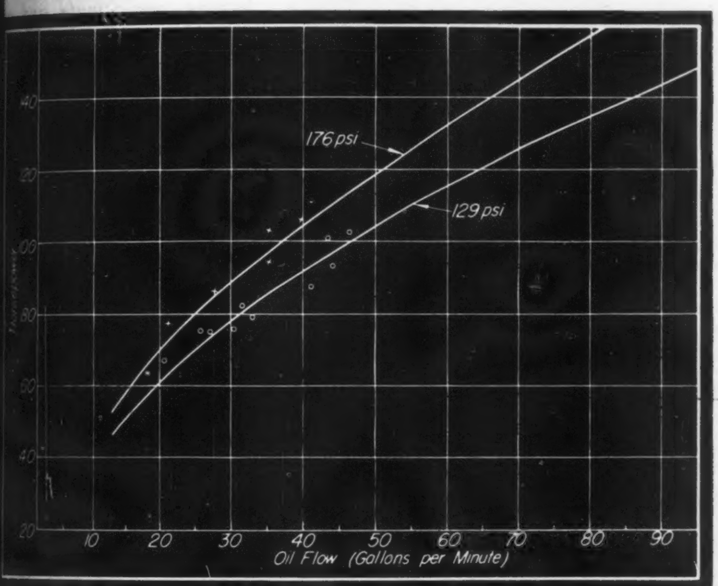
Power losses in the bearings were calculated by measuring the amount of heat added to the oil as it passed through the bearing, and then converting this figure into

orsepower. Radiation losses from the pedestals were eglected because calculations showed them to be small mpared to the total heat generated in large bearings.

Typical test results are shown in *Figs.* 4, 5, 6 and 7, in hich bearing horsepower is plotted as a function of oil w. In *Fig.* 4, two test curves of this type, taken with ifferent bearing loads, are shown. The same oil, oil supy temperature, test speed and the same size bearing ere used in each test so that a direct comparison could e made for the effect of change in load. The test data d the calculated values, shown as solid curves superposed over the test points, indicate that the bearing orsepower is approximately proportional to the square ot of the unit load on the bearing when other operating conditions are not changed.

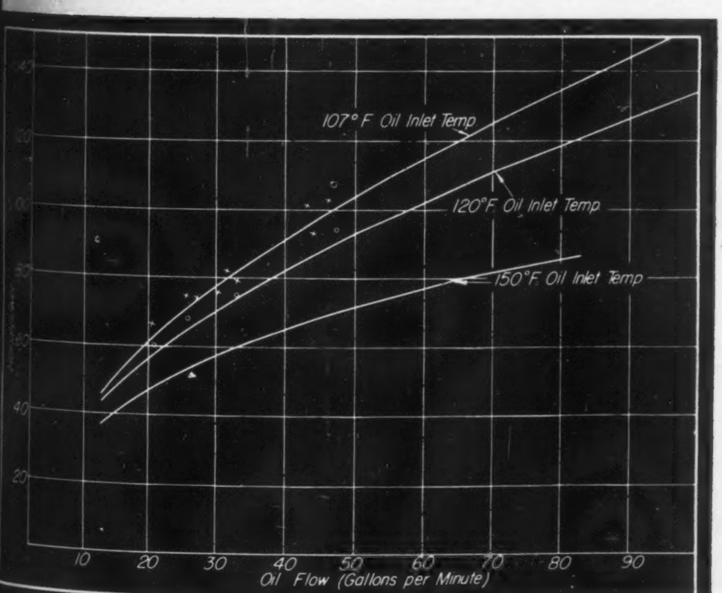
In *Figs.* 5 and 6 are indicated the effect on power losses when the oil viscosity is changed. In *Fig.* 5 the same oil was used for all tests and the viscosity was varied by changing the oil supply temperature. In *Fig.* 6 the oil supply temperature was maintained constant and the viscosity was changed by using two different oils. Examination of these two sets of curves indicates that the power loss curve at a supply temperature of 120 degrees Fahr. for the oil having a viscosity of 210 S.S.U. at 100 degrees Fahr. is approximately the same as the power loss curve at 107 degrees Fahr. for the oil having a viscosity of 150 S.S.U. at 100 degrees Fahr.

This is confirmed by checking the two oils on a standard viscosity chart which shows that the heavier oil has the same viscosity at 120 degrees Fahr. as the lighter oil has at 105 degrees Fahr. In order to keep the same power loss in the bearings when using the heavier oil, it was necessary to increase the oil supply temperature about 15 degrees Fahr. over the temperature used with the lighter oil.



ig. 4—Above—Horsepower loss as function of oil flow for 12-inch iameter bearings with different unit loadings at 3600 revolutions per inute. Oil viscosity 210 S.S.U. at 100°F, supply temperature 107°F

ig. 5—Below—Effect on horsepower loss of oil flow for 12-inch diameter earing carrying 129 pounds per square inch at 3600 revolutions per inute. Oil viscosity 210 S.S.U. at 100°F, inlet temperatures as indicated



Effect of Bearing Top Design

How a change in design of the unloaded portion of a bearing can affect the power loss is shown in *Fig.* 7. One curve shows typical power losses for a 12-inch diameter bearing having a tapered relief on each side but no relief across the unloaded top of the bearing. The other curve shows tests made on the same bearing in the same identical setup after the top of the bearing had been relieved 3/32-inch deep for about 85 per cent of the length. Just making this change in the unloaded part of the bearing and leaving everything else the same reduced the power loss in the bearing by 18½ per cent.

In estimating power losses and oil requirements for turbine bearings, still another problem is to know how to evaluate the oil viscosity. The oil temperature which governs the viscosity is a variable throughout the bearing. This is especially true in large high-speed bearings where large quantities of oil are circulated for cooling purposes.

Finding a Viscosity Criterion

For bearing calculations a viscosity term that can be established by easily measured quantities is desirable. The viscosity for the average of the bearing inlet and outlet temperatures is frequently used, but it was found that this viscosity value did not give results that agreed with tests when the rate of oil circulation was changed. By checking the test data in various ways it was found that when the viscosity was expressed as a ratio of the absolute viscosity at outlet temperature to the square root of the abso-

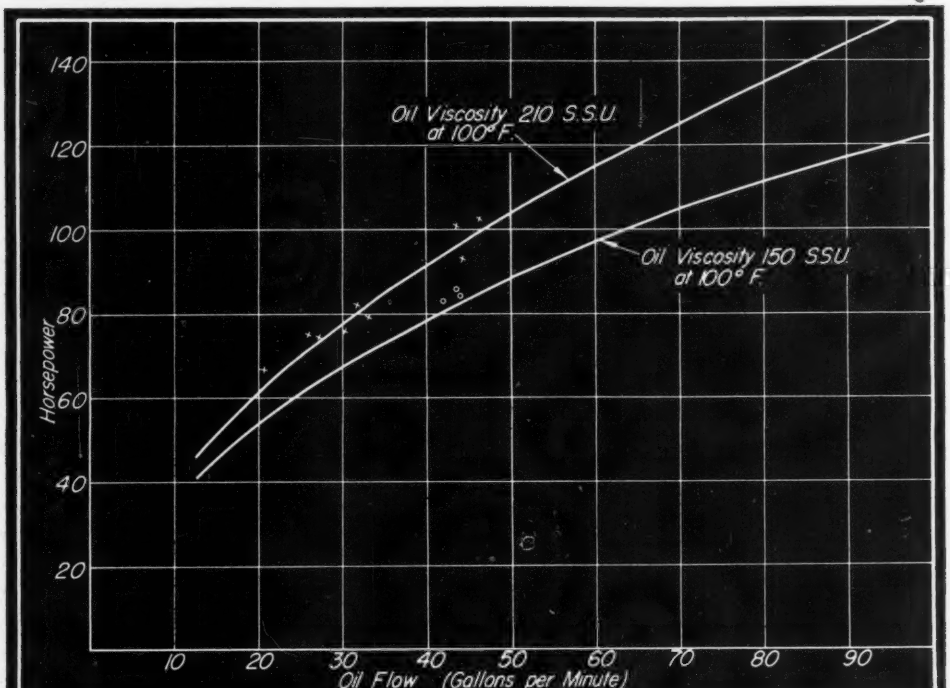


Fig. 6 — Left — Horsepow loss as function of oil fl for different viscosities, usi a 12-inch diameter beari with a load of 129 pounds p square inch, speed 3600 rev lutions per minute and supply temperature 107°

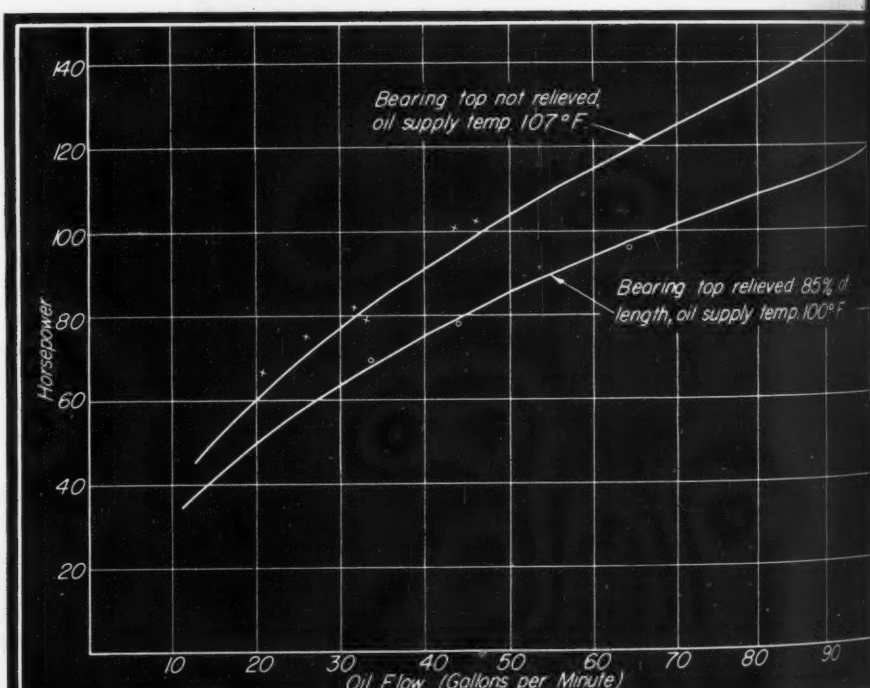


Fig. 7—Right—Effect of design of unloaded portion of bearing on horsepower loss for a 12-inch diameter bearing carrying a load of 129 pounds per square inch at a speed of 3600 revolutions per minute

lute viscosity at inlet temperature, close agreement with all test results could be obtained.

Product of this viscosity ratio and the revolutions per minute, when plotted as a function of the ratio of the coefficient of friction to bearing diameter, gave a straight line on log-log paper. *Fig.* 8 shows curves of this type for bearings with and without top relief. Calculations using these curves gave results that agreed well with the test data, and the curves have proved useful in predicting power losses and oil requirements for other sizes of similarly designed bearings.

The work of using these curves in bearing calculations is simplified by plotting the viscosity ratios for the oil to be used, as shown in *Fig.* 9. When the oil inlet and outlet temperatures are given, the viscosity ratio can be read directly from *Fig.* 9.

The following example illustrates how the curves can be used in estimating bearing performance:

GIVEN:

Bearing diameter d	$= 12$ in.
Bearing effective length l	$= 10\frac{1}{2}$ in.
Bearing unit load p	$= 176$ psi

olutions per minute N	$=3600$
viscosity at 100°F	$=150$ S.S.U.
supply temperature	$=120$°F
half of bearing not relieved	
perature rise of oil passing through bearing	$=30$°F

EQUIRED: Power loss in bearing and oil required, gallons per minute.

OLUTION: From *Fig.* 9, the viscosity ratio 120 degrees Fahr. oil inlet and 150 degrees Fahr. oil outlet is $Z_1/\sqrt{Z_2}=2.3$. Then:

$$\frac{Z_1}{\sqrt{Z_2}}N=2.3\times3600=8280$$

, from Fig. 8, $f/d=.00077$ so that

$$f=12\times.00077=.00924$$

rsepower therefore may be calculated as ows:

$$hp=\frac{pld^2fN\pi}{12\times33000}$$

$$=\frac{176\times10\tfrac{1}{2}\times12^2\times.00924\times3600\times\pi}{396000}$$

$$=70.4$$

After the horsepower has been determined above, oil requirements can be calculated ng the following simplified formula, ich is based on an average value of the cific heat for turbine oil:

$$gpm=\frac{12.5\times hp}{Temp.\ rise}\ \ldots\ldots\ldots\ldots\ldots\ldots(1)$$

ere gpm = oil flow in gallons per minute. this case, required oil flow is:

$$\frac{12.5\times70.4}{30}=29.3$$

Another useful equation, derived from e orifice formula, is helpful in proportion- the leakage area A in the bearing to get desired oil flow gpm for a specified pres- e drop P. This formula can be expressed follows:

$$gpm=8A\sqrt{P}\ \ldots\ldots\ldots\ldots\ldots\ldots(2)$$

e 8 is an empirical constant for turbine , which has been checked by repeated s on many different bearings. The area s the total area in square inches through ich oil can leak out at both ends of the

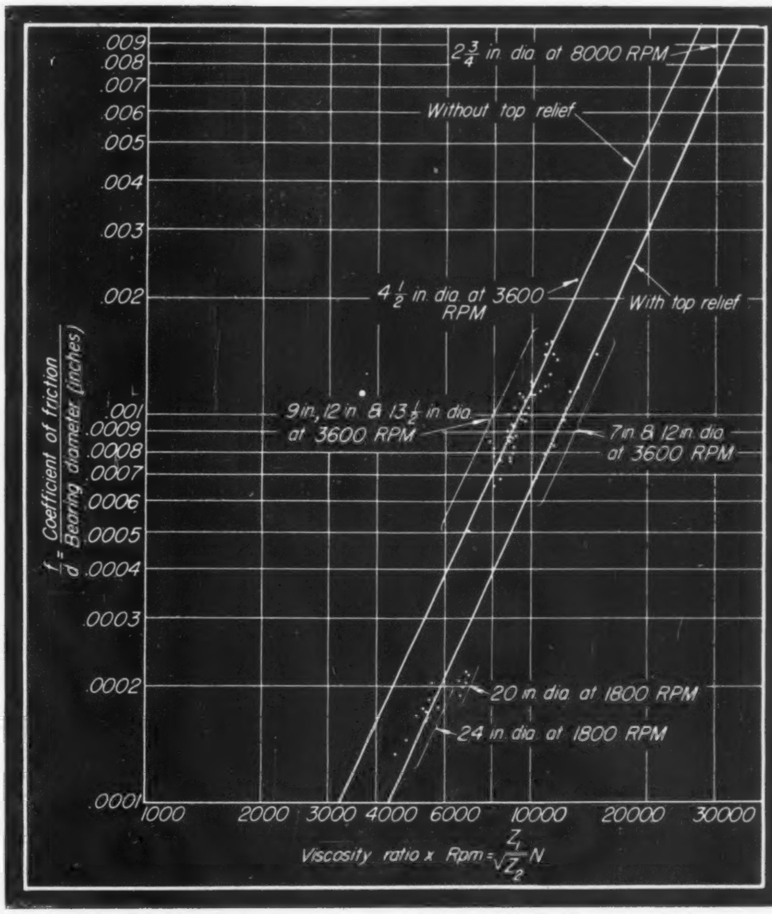
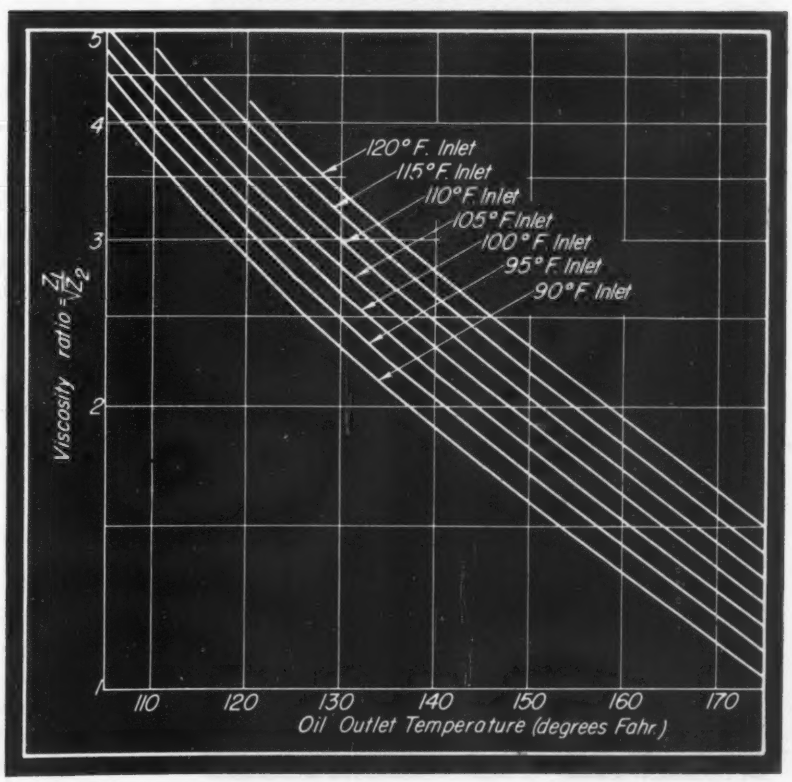


Fig. 8—Above—Chart which can be used to predict power losses in bearings with or without top relief

Fig. 9—Below—Viscosity ratio curves for a turbine oil having a viscosity of 150 S.S.U. at a temperature of 100°F



bearing. Equation 2 can be applied to the problem in the preceding paragraph as follows: For an oil supply pressure of 10 pounds per square inch, the leakage area necessary in the bearing to pass 29.3 gallons per minute is

$$A=\frac{gpm}{8\sqrt{P}}=\frac{29.3}{8\sqrt{10}}=1.16 \text{ sq in.}$$

If the diametral clearance in a 12-inch diameter bearing is .020-inch, the total leakage area between bore of bearing and turn of journal is

$$2\times.010\times12\times\pi=.76 \text{ sq in.}$$

In order to get a total area of 1.16 square inches, it is necessary to provide slots having an area equal to .4 square inches in the unloaded part of the bearing. Two slots at each end, each slot being 3/32-inch deep × 1 1/16-inches long will give the required area.

Performance Chart for a Bearing

An interesting way of presenting bearing data to show the characteristics of a bearing at a glance for a wide range of speeds and temperature rises is shown in *Fig.* 10. For a particular bearing having constant load, constant oil supply temperature, and constant temperature rise, the bearing horsepower varies

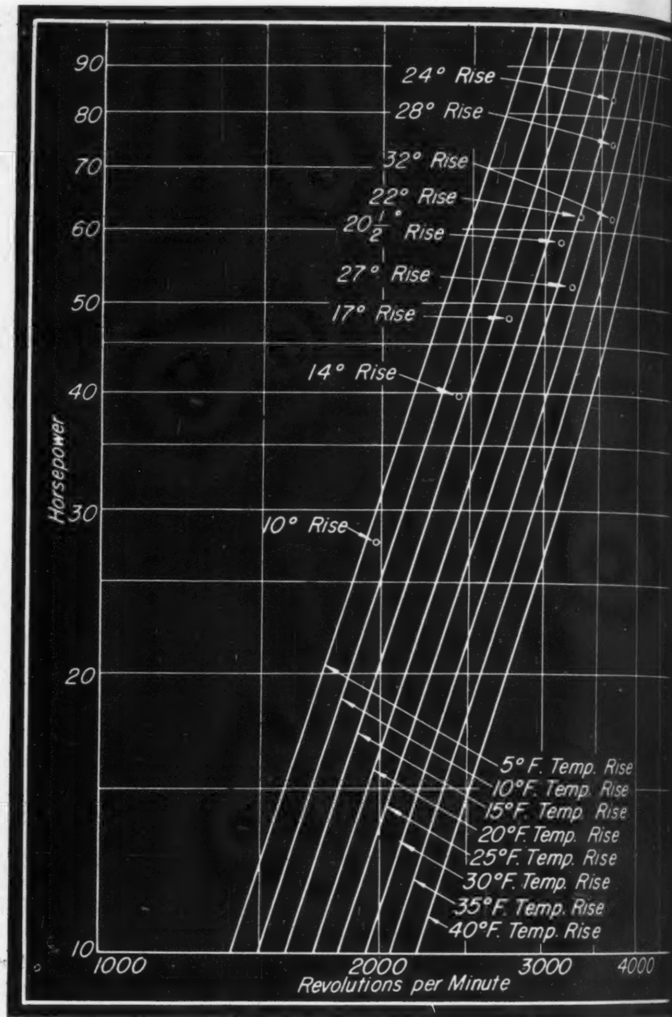


Fig. 10—Above—Operating characteristics of a parti bearing for a wide range of speeds and temperature ris

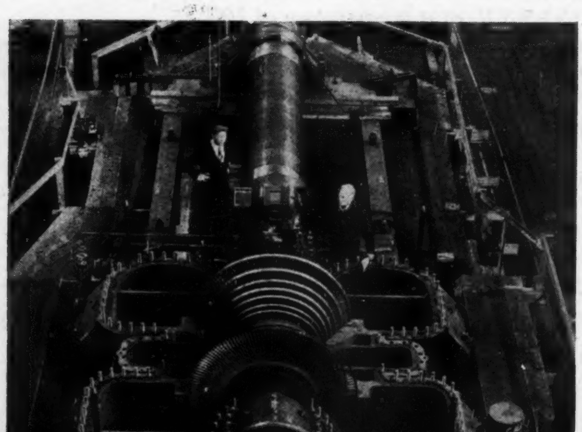
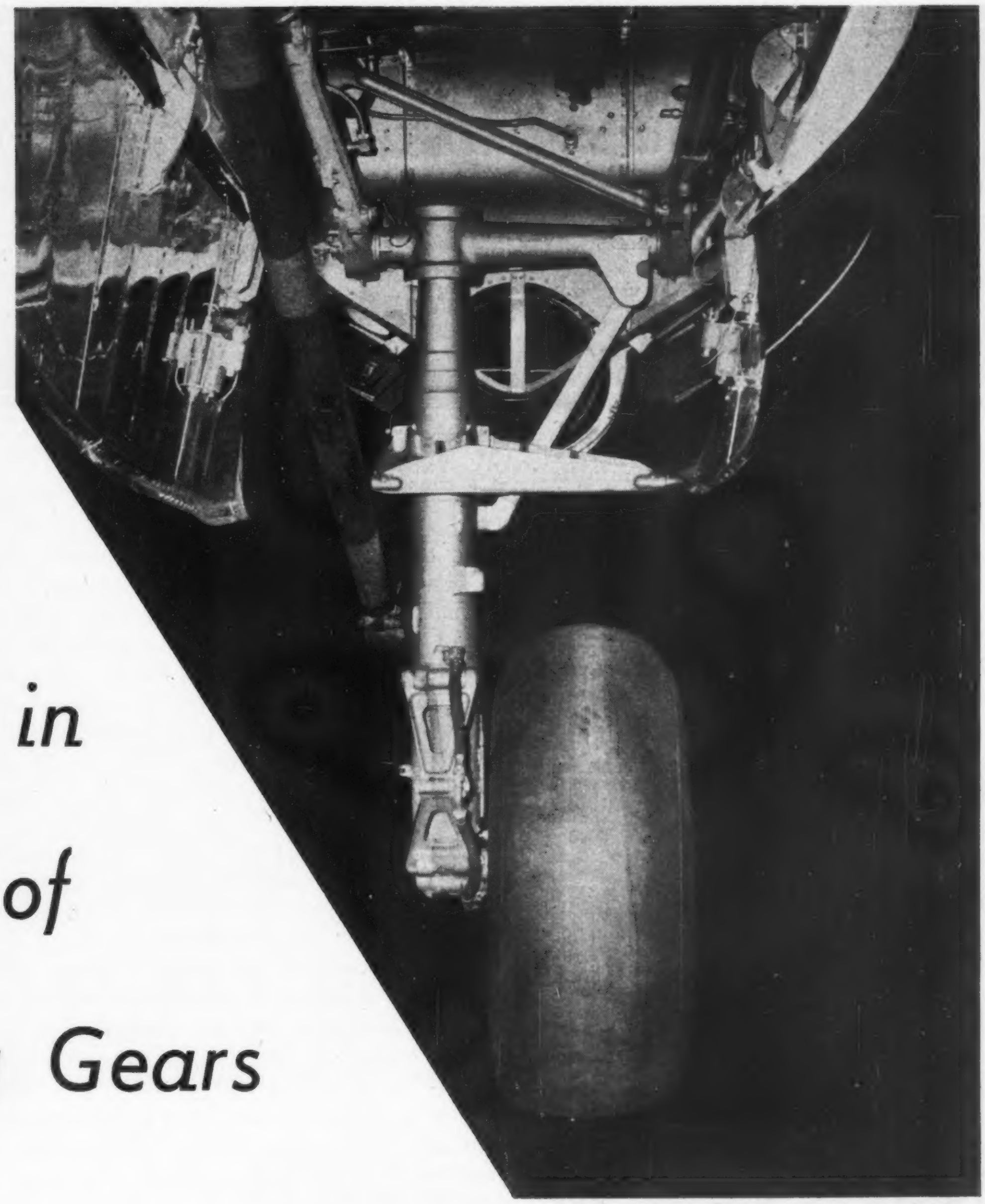


Fig. 11—Left—Bearings for this 25,000-kilowatt turbine w tested with oil flows ranging from 16 to 100 gallons per mi which is considerably above and below normal requirem

as a straight line with revolutions per minute when plotted log-log paper. Using the curves described above, horsep values for two different speeds can be quickly calculated f whole series of temperature rises and a series of curves pl as shown on *Fig.* 10. From this set of curves, the power lo the bearing can be read directly for any desired speed and t perature rise. After the horsepower and temperature rises known, the gallons per minute of oil required can be qui obtained by substituting in Equation 1.

MACHINE TOOL trading pit, devised as a means promptly redistributing production equipment for use in ac erating the manufacture of munitions, has been established the military services and Government production agen Through the relocation of equipment as it becomes available i hoped that the demand on skilled manpower and critical ma rials needed in manufacturing new equipment will be redu



1—Semicantilever ...ing gear of the ...tin A-30 Baltimore ...k bomber. Com- ...gear allows a ...d factor of 6g

actors in esign of anding Gears

By Wilbur A. Taylor

The Glenn L. Martin Company

PROBLEMS involving stress analysis, hydraulics, mechanical functioning, etc., encountered in the design of airplane landing gears are strikingly similar to those posed in the design of mechanisms utilized in ...ny other kinds of machines. For example, any portable machine in ...ich high strength-weight ratios are desirable, presents a fertile field for ...e exploitation of the design principles employed in the development of ...dern landing gears.

Because a landing gear contributes nothing to an airplane's normal ...ction—that is, flying—it must be designed as light in weight as possible ...d still perform its functions as a means for supporting the airplane with... mechanical or structural failure. A typical landing gear of modern de...n is pictured in *Fig.* 1.

Generally speaking, landing gears are of three basic types (see *Fig.* 2). ...e full cantilever gear usually is employed on light aircraft using fixed landing gear or on fighter types where a minimum of retracting space is available for the supporting struts. The semicantilever strut is used most widely on heavy airplanes where the wheels are retracted into wings or engine nacelles. The brace type usually is designed for the fore-and-aft or drag loads, while the side loads are transmitted to the structure of the wing by the trunnion fitting at the top end of the shock strut. Braced gear is used on nonretractable systems such as are used on many light aircraft.

Shock-absorbing landing gear of today consists of an oleo-pneumatic shock absorber in combination with a pneumatic tire. The inside of a typical shock absorber is shown in *Fig.* 3. As the wheel

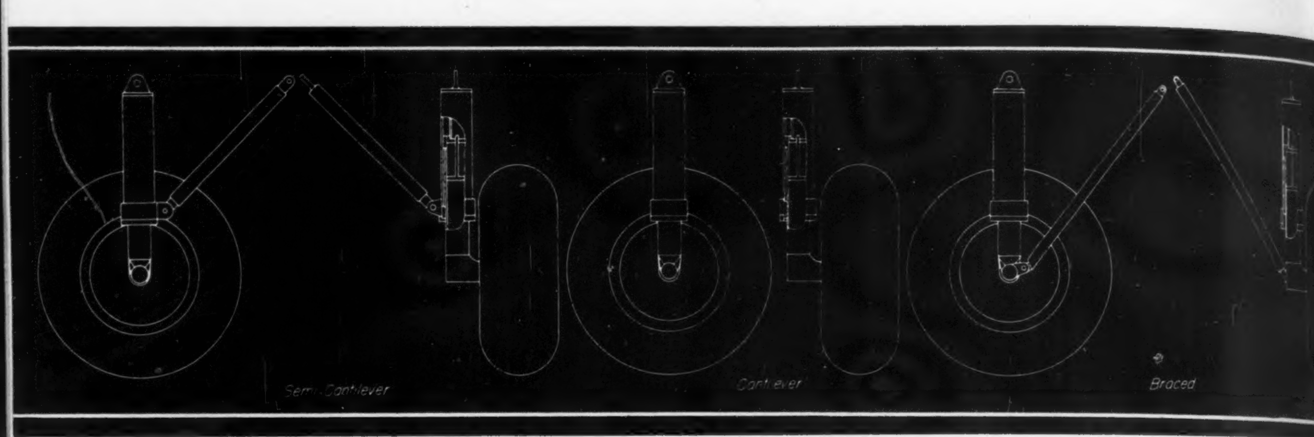


Fig. 2—Above—Basic types of landing gear. Semicantilever type generally is used on heavy aircraft and fixed cantilever gear on light craft. Braced type often is nonretractable

Fig. 3—Below—Typical oleo-pneumatic shock absorber. Oil meters through orifice from chamber 1 to chamber 2

contacts the ground the first action is the flattening of the tire. Then piston is thrust upward into the cylinder, causing the oil in chamber 1 to forced through the orifice around the metering pin and into chamber Since the flow of oil from chamber 1 to chamber 2 is restricted by the ori and metering pin, a hydraulic resistance is set up in chamber 1. This sistance dissipates energy, the dissipation being controlled by the design the metering pin. As the hydraulic fluid flows into chamber 2 it compres the air column which is directly above the oil column. This compression air further dissipates energy by the generation of heat.

It should be pointed out that not nearly 100 per cent of the energy the landing maneuver is dissipated. Much of it is stored in the system, ticularly in the tire, and as soon as the load is off the strut, recoil takes pl Unless corrective means are employed, the airplane will bounce. To gu against this a recoil valve is installed in the cylinder so that, as the strut tends, the returning fluid is slowed up as it passes through the valve.

Although an infinite number of landing attitudes of an airplane are sible, for purposes of stress analysis a limited number are investigated. Th are assumed to impose the most severe loads on the struts. They are: L landing, three-point landing, braked landing, side-drift and one-wh landing.

For semicantilever and cantilever gear, the beam analysis principle followed in which the shock strut is the beam, being loaded in bending a shear and in most cases torsion. The bending between the end of the c inder and the axle may be analyzed—if a forked gear is used—by the curv beam-under-bending method as outlined in reference (1).

When the gross weight of the airplane is known, the required intern air pressure in a landing gear cylinder may be determined from *Fig.* 4. T load on the strut, in the case of the main gear, is one-half the airplane gr weight. These pressures are safe insofar as blowing out the packing se is concerned.

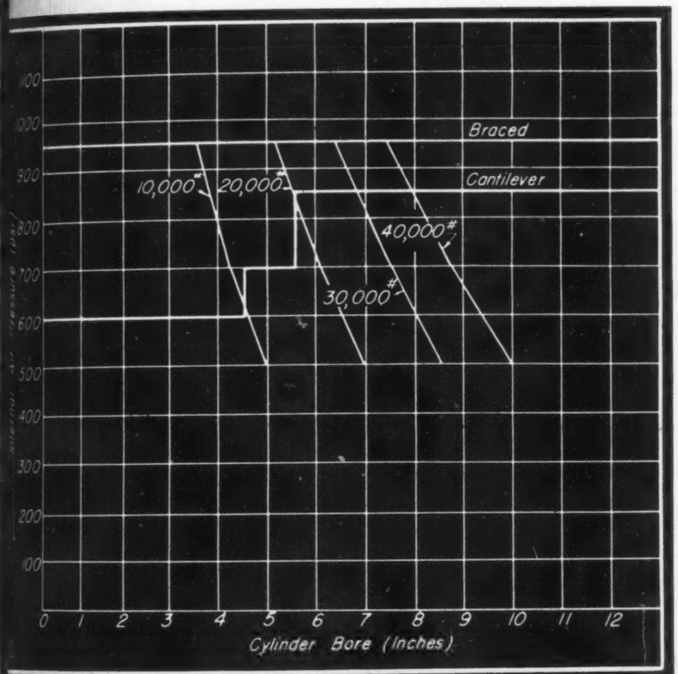
It is standard practice to allow approximately 18 per cent of the pis stroke for taxiing. Since the air column in the cylinder acts during taxiin the proper compression ratio is necessary. This usually is 3 to 1 in ord that 18 per cent of the stroke will be available. Oftentimes this 18 per ce of stroke for taxiing causes the shock absorber to "bottom" (top of the pist strikes the under surface of the cylinder dome), and constant bottomi gives rise to fatigue and consequent failure of the parts. Bottoming is course a function of compression ratio, the higher the ratio the less the p sibility of bottoming. However, it is equally as important not to increa the compression ratio excessively because the basic gas law $P^1V^1=P$ must apply in the design of the shock strut. From this principle it becom clear that a high compression ratio will produce an exceedingly high cylind pressure. The pressure curves in *Fig.* 4 were derived by maintaining a co stant compression ratio, but varying the load factor on the strut. Since t load factor for a large airplane is less than the load factor for a small o the internal cylinder pressures are increased with an increase in airpla gross weight, remaining constant above a certain weight airplane becau

…ction in load factor is not appreciable between two …y airplanes.

…oad L on the strut and the internal air pressure P be… known, the required piston head diameter may be …d by

$$D=\sqrt{\frac{L/P}{.7854}}$$

…y referring to *Fig.* 5 the piston tube diameter corre…nding to the piston head diameter may be selected. …e chart allows for the required thickness of the piston …d bearing and diameter of the piston head is of course



…g. 4—Above—*With these curves, air pressure* … *shock-absorber cylinder is determined*

…entially the same as the cylinder bore with… the tolerance limits. Loading conditions … discussed above are taken into considera…n and from a preliminary stress check, as …tlined in reference (1) or any other reliable …rk on stress analysis, the cylinder wall …ickness may be found.

After selecting the type tire to be used on …e landing gear, and knowing the load on the …re, any tire manufacturer's catalog may be …nsulted. In tabular form will be found the …e diameter required, its inflation pressure, …ergy absorption capacity, etc. With this …ta the stroke of the shock absorber piston …n be calculated by formula. Since several …rmulas exist, each being applicable to any …rplane, the wing-lift formula will be used for …ustrative purposes. This assumes a large …rt of the airplane to be air-borne at the in…nt of impact. It also assumes that this …-borne part is the same for all airplanes no …atter what the aerodynamic characteristics of the airplane may be. The wing-lift formula is as follows:

$$S=\frac{E_a-E_t}{\eta(L.L.F.-1)W}$$

Where:

E_a = ½ MV^2 or WH
H = drop height
E_t = energy absorbed by tire × .60
W = ½ airplane gross weight
η = shock strut efficiency (usually 75 per cent)
$L.\ L.\ F.$ = limit load factor
V = vertical rate of descent of aircraft at point of contact with ground.

All of the basic dimensions of the shock absorbing system can be determined from the above data. The length of the shock strut is determined by the required propeller clearance and retraction conditions. A strut may be long enough to give the required angle of attack for take-off but it may also interfere with some major structural component of the wing or fuselage. If such a condition should exist, it is of course self evident that compromises must be made between the stress analyst and engineer responsible for the wing or fuselage design, the aerodynamists and the landing gear designer. This parallels the kind of close cooperation necessary between research, development, design and production engineers throughout the entire machine building industry if progress is to be realized in its fullest sense in the future.

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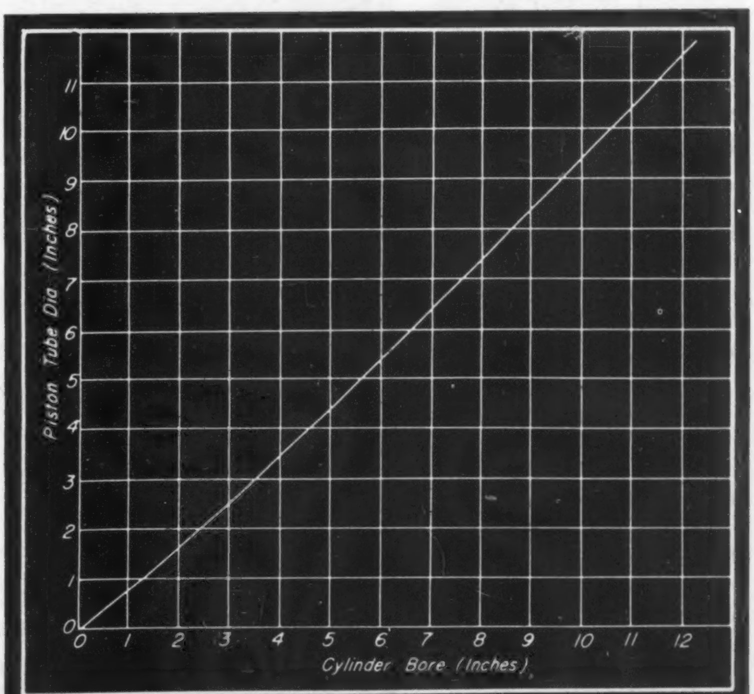
Fig. 5—Below—Curve gives corresponding diameters of piston tubes and heads



Develops Power Stroke with Boosters

By B. D. Johnson

The C. A. Lawton Co.

HIGH-PRESSURE work stroke for a hydraulic press for drawing shell is developed through two double-acting booster units as indicated in the accompanying schematic circuit. Primary power is obtained in the press from two variable-volume pumps with automatic pressure governors delivering oil at 750 pounds per square inch, while the secondary power for the work stroke, developed by the booster units, provides a pressure of 3000 pounds per square inch.

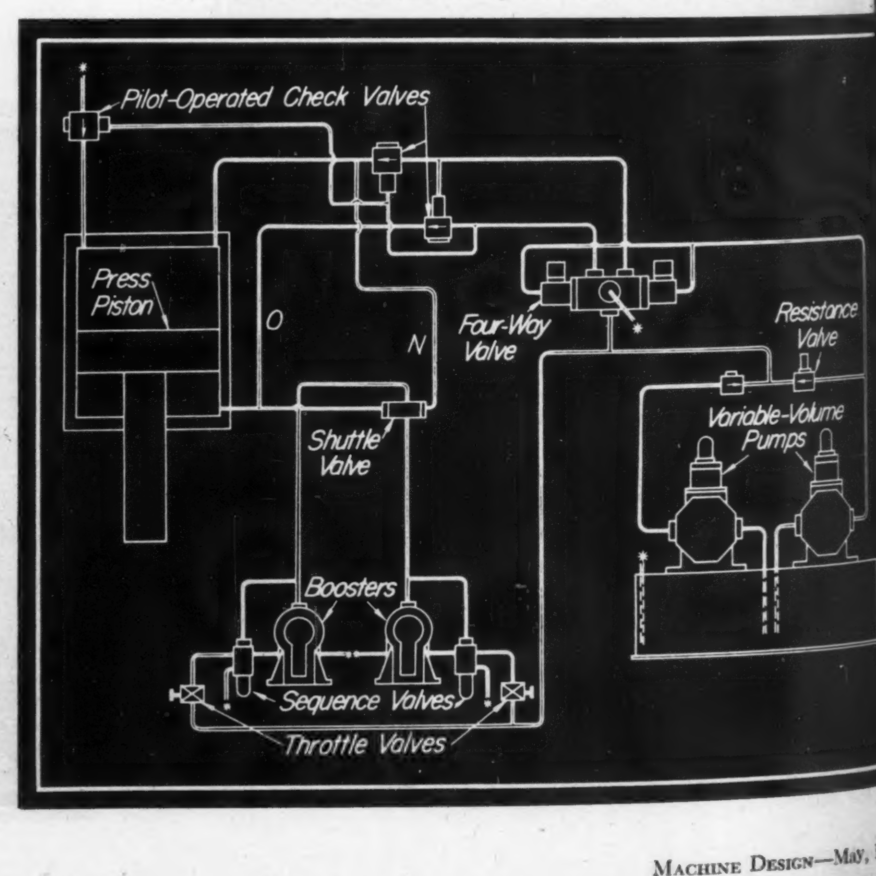
Direction control for the press utilizes a standard spool-type, four-way, solenoid-operated valve. Operation of a pushbutton shifts this valve to apply pressure to the top of the press piston. Flow from both pumps advances the piston at rapid traverse. The pressure on line *N* places a shuttle valve in position to open both boosters' output to the line. When the press platen meets work resistance, sufficient pressure is developed to open two sequence valves, starting intensification of pressure through the boosters. Combined flow of the boosters exerted through the shuttle valve advances the press platen at high pressure to draw the shell. To prevent reverse flow of the secondary pressure through the four-way valve, a check valve is utilized.

Pressure Reverses Valve

Completion of the draw stroke and development of maximum pressure reverses the four-way valve through a pressure switch, applying primary pump pressure to line *O*, which again reverses the shuttle valve and applies booster pressure to the bottom of the cylinder giving high tonnage for withdraw punch from the work.

Features of this hydraulic circuit include: Mo driven pumps and solenoid-operated four-way valve subject to only moderate pressures. High pressure work stroke is developed by 4 : 1 intensification of pu pressures through hydraulic boosters. Output of boos is valved automatically to either side of the cylin through a shuttle valve. Exhaust oil on upward stroke ram is vented direct to tank through a pilot-operated ch valve. One variable-delivery pump is vented to t when the four-way valve is in neutral. The other pump maintained at low pressure which is set by a resista valve to maintain low pilot pressure for operation of fo way valve. Speed of draw stroke is steplessly vari through throttle valves. Pilot-operated check valves p vent high-pressure oil from reaching the solenoid-opera four-way valve.

Schematic of hydraulic circuit for press for drawing shell. Moderate pressure of 7 pounds per square inch is utilized for primary power while boosters intensify press four times for work stroke during the deep-drawing operation



What the Veteran Offers

RETURN of American prisoners of war and of increasing numbers of service men from battle zones and training camps brings into sharp focus the question of re-employment in civilian life. Too much emphasis cannot be placed on this subject now in view of the individual hardships and the national situation that will arise if it is neglected until too late.

Much of the consideration, however, so far devoted to this phase of the war and its aftermath seems to lie in the direction of "doing something for the veteran". Is it not time that this thinking, praiseworthy as it is, be changed so that the viewpoint of the vast majority of veterans also be taken into consideration and their potential value on return to civilian occupations be recognized?

Many of these men have qualifications to offer that have not yet been taken greatly into account. Some of those who joined the services with little or no experience in industry have had the opportunity to gain valuable knowledge in technical branches of the Forces; others who were relatively young at the time of induction will have broadened their vision to the point of being capable of accepting considerable responsibility and, in many cases, active leadership; and still others—especially those with lengthy overseas experience—will be in a position to apply their knowledge of conditions abroad to the development and production of goods most adaptable to foreign markets.

Veterans handicapped to a greater or lesser degree by wartime injuries might well be considered on this same basis rather than from a beneficent standpoint. Amputation cases, taken as an example, are estimated to involve about one per cent of the total of those wounded. Such cases present difficulties but these should be readily surmountable in most instances. All these men want is to be given the opportunity to prove their worth in work that does not fall too far beyond the range of their capacities.

L. E. Jermy

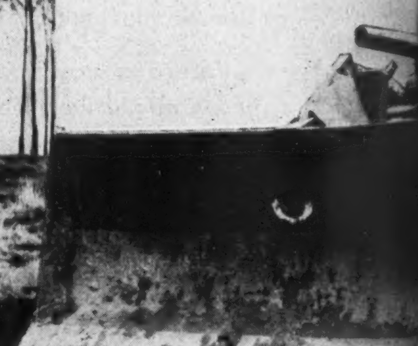
M-10 Three-Inch Gun Motor Carriage (Left). Twin six, two-cycle diesel, water-cooled engine; carries 3-inch gun in power turret; max speed, 25 mph; cruising radius, 200 miles; can climb 50 per cent grade; carries two-way radio

Fighting Machines of America

Herein pictured are some of the world's foremost machines of war. They're now on the high road to Berlin and Tokyo —and total victory!

M-29C "Water Weasel" (Below). Powered by six-cylinder Champion engine; 20-inch wide tracks driven from rear axle through sprockets; bogie wheels mounted in clusters of four on transverse leaf springs; amphibious personnel and materiel carrier; negotiates rough terrain as well as water

Tankdozer (Below). M-4 tank with bull-dozer equipment attached; 30-cylinder liquid-cooled engine; 75-mm rifle, 50 and 30-cal. machine guns; max speed, 25 mph; cruising radius, 160 miles; will climb 27 per cent grade



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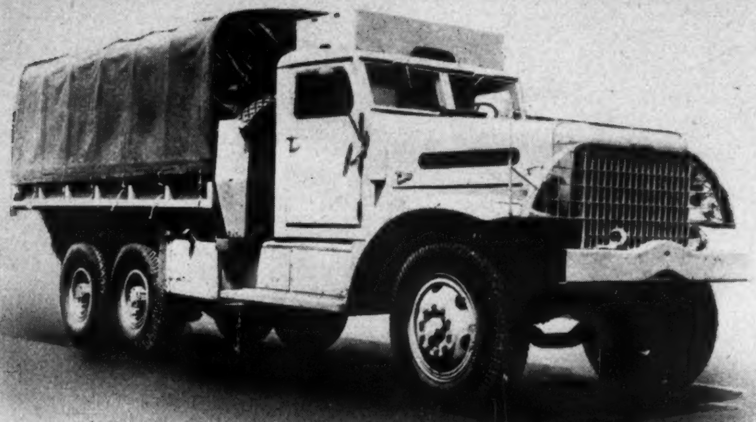
M-24 Combat Tank (Below). Automatic drive through eight speeds forward, four reverse; powered by two V-type Cadillac, 8-cylinder engines; important armored surfaces sloped at 45 degrees; mounts 75-mm cannon, 50-cal. antiaircraft gun, smoke mortar, 30-cal. coaxial gun, and 30-cal. bow machine gun; 16-inch wide tracks effect low ground pressure; can climb 60 per cent grade

M-8 Light Armored Car (Above). Six-cylinder, water-cooled Hercules engine, 110 hp; drives through all wheels; max rated speed, 55 mph; cross-country cruising radius, 100 to 200 miles; on highway, 200 to 400 miles; can climb 60 per cent grade; 37-mm gun in turret and 30-cal. machine gun

M-18 Tank Destroyer "Hellcat" (Right). Nine-cylinder, single-row, radial, aircraft-type engine, 480 hp; rated at 55 mph max speed; weight, 19 tons; mounts 76-mm cannon, also machine guns; centrifugal blowers used for oil cooling; track comprises 83 heavy drop-forged links, hard-faced to reduce wear

M-1 Heavy Wrecking Truck (Below). 6-cylinder in-line engine; lifting capacity, 10,000 lbs; max speed, 45 mph; cruising range 200 miles; equipped with air brakes

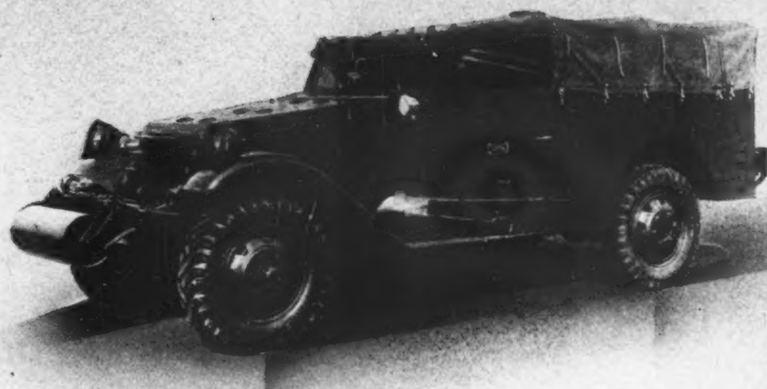
M-1 Rocket Launcher "Bazooka" (Be-
low). Electrical firing set off by
trigger uses dry cell batteries; effec-
tive range, 200 to 300 yds



Ten-ton "Six by Four" Cargo Truck (Above). Six-wheeled; drive through four wheels; double reduction, full floating axles; five speeds forward, one reverse; all-steel body 180 inches long, 88 inches wide; 200-inch wheelbase; air brakes; can climb 32 per cent grade, fully loaded

Four-Forty-Four Truck Tractor (Right). Drive through all four wheels; cab-over-engine model; 134½-inch wheelbase; eight speeds forward, two reverse; used to haul gas-tank trailers in cross-country and airport refueling operations

M-16 Machine Gun Turret (Below). Power driven in azimuth and elevation; mounts four 50-cal. machine guns fired synchronously through solenoid switches; V-belt variable speed drive powered by electric motor; drive under control of double pistol grip; has been used with great success against enemy dive bomber attacks



Scout Car (Above). Drive through all four wheels; eight speeds; two self-sealing fuel tanks with total capacity of 60 gallons; cruising range, 350 miles; has two-way radio; armor-plate shutters over radiator; roller in front aids in getting out of holes and ditches

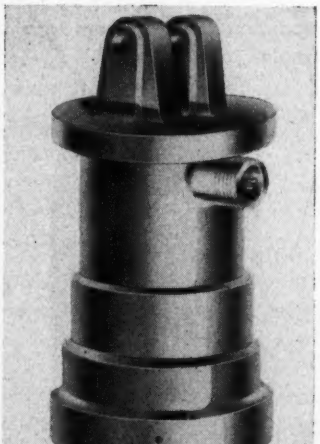
Diesel-Hydraulic Fork Lift Truck (Right). Capacity, 3 tons; hydraulic tilting mechanism permits backward and forward tilting of the lifting mast. Used by Navy in unloading barges and moving materials into storage and dispersal areas

40-mm Bofors Gun (Below). Antiaircraft and antitank; max vertical range, approx. 7600 yds; horizontal range, about 11,000 yds; rate of fire 120 rounds per minute max; fired by foot-pedal control; fires either high explosive or armor piercing shell; weight 5549 lbs

Airport Fire Truck (Below). Six-wheeled; all-wheel drive; air brakes; 180 to 200-hp engine; fifteen-foot nozzle-boom at top and front nozzle controlled by pistol-grip controls in cab; ground-sweep nozzles below front bumper

Applications

of Engineering Parts, Materials and Processes



Actuator Design Facilitated

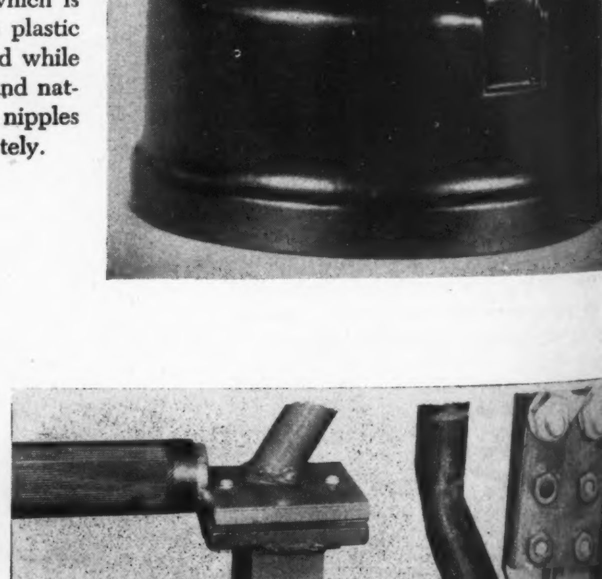
AIRCRAFT actuator shown at left incorporates the Saginaw Steering Gear division ball bearing screw and nut such as is used on GMC trucks. Employed to actuate doors, landing gears, control surfaces, etc., the unit offers a number of desirable features including low friction and positive mechanical action. Compact and light in weight it is readily adaptable to emergency hand operation should a power failure occur.

Insulator Life Is Improved

MATERIALS for application as automotive distributor cap nipples, right, in addition to elasticity and good insulating properties must possess resistance to ozone. The illustration shows what happens to various materials after subjection to highly concentrated ozone for 6 hours. The nipple at left of the picture, which is made of an elastomeric plastic (Vinylite), is unimpaired while the synthetic (center) and natural rubber (right) nipples cracked or failed completely.

Speeds Automatic Welding

UTILIZING direct current, bare electrodes and a granular flux which completely blankets the arc and molten metal, the Lincolnweld process of automatic arc welding is claimed to be simpler, faster and more economical than previous procedures. In the illustration at right the equipment is shown applied to the welding of a butt joint. Just behind the roller guides which engage the prepared seam is the tube which feeds flux from a hopper. Behind the tube is the electrode which is fed from a reel at a controlled rate. Less sensitive to scale and moisture, the process dispenses with rigorous cleaning operations prior to welding.



Improved Methods for Calculating Torsional Vibration

By Robert H. Scanlan
Chief Stress Analyst
Lawrance Aeronautical Corp.

Part 2—Higher Modes

IN PART I of this series, published last month, the general method of applying matrix techniques to the solution of torsional vibration natural frequencies and modes was developed. A numerical example, covering a 4-cylinder internal combustion engine with flywheel, was calculated for the fundamental mode. The mode, or normal elastic curve, worked out from the example, may be plotted from TABLE IV. For calculating the higher modes, three possible methods may be employed. These are discussed in the following.

First method of obtaining higher modes makes use of the *orthogonality relation* between the coordinates ϕ of any two distinct modes. This condition depends on the fact that perfect elasticity holds throughout the shaft system and is a representation of the fact that any motion is the sum of separate and distinct harmonics. The relation is not derived here, but merely stated, as follows (7):

$$I_0\phi_0\phi_0'+I_1\phi_1\phi_1'+I_2\phi_2\phi_2'+I_3\phi_3\phi_3'+I_4\phi_4\phi_4'=0 \quad \ldots\ldots\ldots (9)$$

where the ϕ_0, ϕ_1, ϕ_2, ϕ_3, ϕ_4 are the coordinates of any mode distinct from ϕ_0', ϕ_1', ϕ_2', ϕ_3', ϕ_4', the coordinates of another mode. By using Equation 9 together with Equations 7 and 5 the iteration process may be made to yield the second mode. Once the second mode has been obtained, use of Equation 9 twice, expressing the orthogonality between first and third modes and between second and third modes, yields the necessary condition on Equation 7 and 5 to obtain from it the third mode by iteration. For each new mode sought, Equation 9 must be used an additional time. It is evident that this method becomes cumbersome as higher modes than the second are sought. It is therefore used on the present example to obtain the second mode only.

For the second mode, Equation 9 is

$$600(-.4604)\phi_0+100(.2818)\phi_1+100(.6150)\phi_2+$$
$$100(.8657)\phi_3+100(1)\phi_4=0$$

or

$$-276.24\phi_0+28.18\phi_1+61.5\phi_2+86.57\phi_3+100\phi_4=0 \quad \ldots [c]$$

*References in parentheses are listed at end of article.

The momentum condition, Equation 5, is

$$600\phi_0+100(\phi_1+\phi_2+\phi_3+\phi_4)=0 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots [d]$$

Eliminating ϕ_0 between [c] and [d] results in

$$1113.3\phi_1+1613.1\phi_2+1989.15\phi_3+2190.6\,\phi_4=0 \quad \ldots\ldots [e]$$

Solving for ϕ_2 from [e] gives

$$\phi_2=-.69016\phi_1-1.23312\phi_3-1.35801\phi_4 \quad \ldots\ldots\ldots\ldots [f]$$

In the iteration process on the series of equations at the top of the first column, Page 162, in Part I, an initial mode satisfying [e] now is chosen. Thus, arbitrarily assuming $\phi_1=-1.0000$, $\phi_3=-.1000$ and $\phi_4=1.0000$, the corresponding assumed value of ϕ_2, calculated from [f], is found to be $-.5445$. Using these figures, the iteration process by column postmultiplication gives the values in TABLE III. It is to be noted that ϕ_2 is not calculated by the matrix multiplication but by [f], to insure use of the orthogonality condition at every step. Thus the matrix multiplication to get the second column in TABLE III from the first is

$$\begin{bmatrix} 1.2 & .9 & .7 & .6 \\ 1.2 & 1.9 & 1.7 & 1.6 \\ 1.2 & 1.9 & 2.7 & 2.6 \\ 1.2 & 1.9 & 2.7 & 3.6 \end{bmatrix} \begin{bmatrix} -1.0000 \\ -\ .5445 \\ -\ .1000 \\ 1.0000 \end{bmatrix} = \begin{bmatrix} -1.1600 \\ \\ -\ .0954 \\ 1.0954 \end{bmatrix}$$

where

$$-1.1600=(1.2\times-1.0000)+(.9\times-.5445)$$
$$+(.7\times-.1000)+(.6\times1.0000),$$
$$-\ .0954=(1.2\times-1.0000)+(1.9\times-.5445)$$
$$+(2.7\times-.1000)+(2.6\times1.0000), \text{ etc.}$$

To get Col. 3, the figures in Col. 2 are normalized, thus $-1.1600/1.0954=-1.0590$, $-.0954/1.0954=-.0871$, and $1.0954/1.0954=1.0000$. Substituting these values of ϕ_1, ϕ_3, and ϕ_4 in [f], ϕ_2 is found to be $-.5197$. The remaining columns are calculated in similar manner until alternate columns show negligible change. To get ϕ_0, the results in the final column of TABLE III are used in Equation 5, giving $\phi_0=+.1234$.

Frequency equation is $10^5/\omega_2{}^2=1.203$, from which $\omega_2=288.3$ radians per second and the frequency $f_2=(60/2\pi)\omega_2=2753$ cycles per minute.

Second method to be described for obtaining higher frequencies is a method of elimination used by the Air Technical Service Command, Wright Field (5). However, while yielding the higher frequencies it does not conveniently yield the corresponding modes. Like the

previous method, it also employs the principle that the motion in any single mode is independent of the motion in any other mode. Thus, if ϕ_n is the total motion of one coordinate during vibration, it may be expressed as a linear combination of the contribution $\phi_n^{(\prime)}$ of the fundamental mode plus the total contribution ϕ_{n0} of higher modes:

$$\phi_n = \phi_n^{(\prime)} + \phi_{n0} \qquad \text{(10)}$$

If, for example, there are four coordinates as in Equation 8*a*, and the fundamental mode ($\phi_1^{(\prime)}$, $\phi_2^{(\prime)}$, $\phi_3^{(\prime)}$, $\phi_4^{(\prime)}$) is known, together with its frequency Ω_1, the introduction of this mode and frequency into Equation 8*a* will give an identity. Each coordinate of the fundamental mode may be expressed as a fraction of a reference coordinate, say $\phi_1^{(\prime)}$, thus:

$$\phi_n^{(\prime)} = c_n \phi_1^{(\prime)} \qquad \text{(11)}$$

Then the identity, Equation 8*a*, gives

$$\left.\begin{aligned}
\frac{1}{\Omega_1^2} &= b_{11}+b_{12}c_2+b_{13}c_3+b_{14}c_4 \\
\frac{c_2}{\Omega_1^2} &= b_{21}+b_{22}c_2+b_{23}c_3+b_{24}c_4 \\
\frac{c_3}{\Omega_1^2} &= b_{31}+b_{32}c_2+b_{33}c_3+b_{34}c_4 \\
\frac{c_4}{\Omega_1^2} &= b_{41}+b_{42}c_2+b_{43}c_3+b_{44}c_4
\end{aligned}\right\} \qquad \text{(12)}$$

From Equations 10 and 11

$$\phi_n = c_n \phi_1^{(\prime)} + \phi_{n0} \qquad \text{(13)}$$

Substituting values of ϕ_1, ϕ_2, ϕ_3, and ϕ_4 from Equation 13 into Equation 8*a* gives a series of equations of which the following is typical:

$$c_2\phi_1^{(\prime)} + \phi_{20} = (b_{21}+b_{22}c_2+b_{23}c_3+b_{24}c_4)\phi_1^{(\prime)}\omega^2 + (b_{22}\phi_{20}+b_{23}\phi_{30}+b_{24}\phi_{40})\omega^2$$

Comparing this equation with the corresponding line in Equation 12, it is evident that it may be written as indicated in the following, which includes the complete series of equations resulting from the foregoing steps:

$$\left.\begin{aligned}
\phi_1^{(\prime)} &= \frac{1}{\Omega_1^2}\phi_1^{(\prime)}\omega^2 + (b_{12}\phi_{20}+b_{13}\phi_{30}+b_{14}\phi_{40})\omega^2 \\
c_2\phi_1^{(\prime)}+\phi_{20} &= \frac{c_2}{\Omega_1^2}\phi_1^{(\prime)}\omega^2 + (b_{22}\phi_{20}+b_{23}\phi_{30}+b_{24}\phi_{40})\omega^2 \\
c_3\phi_1^{(\prime)}+\phi_{30} &= \frac{c_3}{\Omega_1^2}\phi_1^{(\prime)}\omega^2 + (b_{32}\phi_{20}+b_{33}\phi_{30}+b_{34}\phi_{40})\omega^2 \\
c_4\phi_1^{(\prime)}+\phi_{40} &= \frac{c_4}{\Omega_1^2}\phi_1^{(\prime)}\omega^2 + (b_{42}\phi_{20}+b_{43}\phi_{30}+b_{44}\phi_{40})\omega^2
\end{aligned}\right\} \qquad \text{(14)}$$

It is obvious from the form of Equation 14 that the coordinate $\phi_1^{(\prime)}$, can immediately be eliminated from the equations by use of the first equation in the second, third and fourth equations. The result is a set of three equations from which the fundamental mode has been eliminated. These can then be expressed in matrix form and iterated to give the second frequency, as demonstrated in the worked example.

To obtain the next higher frequency at each stage the same technique as described in the foregoing is applied to the last set of equations used for iteration, obtaining a new set with one less equation. At each stage Equation 5 may be used to find ϕ_0. This method is used in the following to obtain the second natural frequency of the system in the worked example, which should (and does) coincide with the result already obtained. In the first mode it was found (TABLE II) that

$$\phi_2^{(\prime)} = \frac{.6150}{.2818}\phi_1^{(\prime)} = 2.1824\phi_1^{(\prime)}$$

$$\phi_3^{(\prime)} = \frac{.8657}{.2818}\phi_1^{(\prime)} = 3.0720\phi_1^{(\prime)}$$

$$\phi_4^{(\prime)} = \frac{1.0000}{.2818}\phi_1^{(\prime)} = 3.5486\phi_1^{(\prime)}$$

Using Equation 13, the following may be written:

$$\phi_2 = 2.1824\phi_1 + \phi_{20}$$

$$\phi_3 = 3.0720\phi_1 + \phi_{30}$$

$$\phi_4 = 3.5486\phi_1 + \phi_{40}$$

Substituting these values in the series of equations at the top of the first column, Page 162, Part I,

$$\left.\begin{aligned}
\phi_1 &= [7.44372\phi_1 + .9\phi_{20} + .7\phi_{30} + .6\phi_{40}]\omega^2\times10^{-5} \\
2.1824\phi_1&+\phi_{20} \\
&= [16.24672\phi_1 + 1.9\phi_{20} + 1.7\phi_{30} + 1.6\phi_{40}]\omega^2\times10^{-5} \\
3.0720\phi_1&+\phi_{30} \\
&= [22.86732\phi_1 + 1.9\phi_{20} + 2.7\phi_{30} + 2.6\phi_{40}]\omega^2\times10^{-5} \\
3.5486\phi_1&+\phi_{40} \\
&= [26.41592\phi_1 + 1.9\phi_{20} + 2.7\phi_{30} + 3.6\phi_{40}]\omega^2\times10^{-5}
\end{aligned}\right\}$$

These equations correspond to Equation 14. To eliminate ϕ_1, both sides of the first equation may be multiplied by 2.1824 and subtracted from the second equation, then both sides of the first equation multiplied by 3.0720 and subtracted from the third equation, etc. As a result three equations are obtained, which may be expressed in matrix form as follows:

$$\begin{bmatrix}\phi_{20}\\ \phi_{30}\\ \phi_{40}\end{bmatrix} = \omega^2 \begin{bmatrix} -.06416 & .17232 & .29056 \\ -.86480 & .54960 & .75680 \\ -1.29374 & .21598 & 1.47084 \end{bmatrix} \times \begin{bmatrix}\phi_{20}\\ \phi_{30}\\ \phi_{40}\end{bmatrix} \times 10^{-5} \qquad \text{(A)}$$

TABLE III

Calculation of Relative Vibration Amplitude in Second Mode

Assumed mode	Col. 2	Col. 3	Col. 4	Col. 5	Col. 6	Col. 7	Col. 8	Col. 9	Col. 10	Col. 11	Col. 12	Col. 13	Col. 14	Mode ϕ
—1.0000	—1.1600	—1.0590	—1.1995	—1.0840	—1.2827	—1.0794	—1.3120	—1.0965	—1.3240	—1.1029	—1.3290	—1.1056	—1.3310	—1.1064
— .5445	. .	— .5197	. . .	— .7215	. . .	— .8085	. . .	— .8037	. . .	— .8028	. . .	— .8022	. . .	— .8024
— .1000	— .0954	— .0871	.1066	.0963	.1884	.1585	.1965	.1642	.2005	.1670	.2021	.1681	.2030	.1687
1.0000	1.0954	1.0000	1.1066	1.0000	1.1884	1.0000	1.1965	1.0000	1.2005	1.0000	1.2021	1.0000	1.2030	1.0000

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Iterating $[h]$ gives, in the final columns:

$$\begin{matrix} .39428 & .3276 \\ .87146 & .7240 \\ 1.20360 & 1.0000 \end{matrix}$$

Then $10^5/\omega_2{}^2 = 1.2036$, $\omega_2 = 288.2$ radians per second, and $f_2 = (60/2\pi)\,\omega_2 = 2752$ cycles per minute.

Third method to be discussed is one which accomplishes essentially the same results as the other two methods, by matrix computation exclusively (4), reducing to routine computation all the work involved. No attempt will be made to give the underlying theory, but the technique will be described in sufficient detail to enable the designer to follow each step.

To obtain the second mode from Equation 8*b*, assume B_1 is the square matrix on the right side of this equation. Let

$$[r_1 \quad r_2 \quad r_3 \quad r_4]$$

be the row matrix obtained by iteration on B_1, by *row* premultiplication (see Part I, Page 161). Assume this is "normalized" by dividing the results through by any one of the r's, say r_i. Then the row has a 1 in the ith place and has the form

$$\left[\frac{r_1}{r_i} \quad \frac{r_2}{r_i} \quad \frac{r_3}{r_i} \quad \frac{r_4}{r_i}\right]$$

Let I be the "identity matrix" with unity on the main diagonal and zeros elsewhere:

$$\begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} = I$$

Let E_1 be the square matrix with the normalized row as its ith row and zeros elsewhere; if $r_i = r_4$, for example, there E_1 is

$$\begin{bmatrix} 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ \frac{r_1}{r_4} & \frac{r_2}{r_4} & \frac{r_3}{r_4} & \frac{r_4}{r} \end{bmatrix} = E_1$$

Form the matrix $I - E_1$; thus, if r_4 equals r_i,

$$\begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ \frac{-r_1}{r_4} & \frac{-r_2}{r_4} & \frac{-r_3}{r_4} & 0 \end{bmatrix} = I - E_1$$

Form the square matrix $B_2 = B_1\ (I - E_1)$ by multiplication. The matrix B_2 will now be used exactly as B_1 was used, in the iteration process, and the result (by postmultiplication) will converge on the second mode and frequency ω_2. It may be noted that iteration by premultiplication may be used if only the frequency is desired, this yielding no information about the mode, however. The process may be repeated to yield the

TABLE IV
Characteristic Modes for 5-Mass System

Mode	1st	2nd	3rd	4th
ϕ_0	—.4604	.1234	.0447	.0127
ϕ_1	.2818	—1.1064	—1.1110	—.5130
ϕ_2	.6150	—.8024	.7070	1.0000
ϕ_3	.8657	.1687	1.0000	—.9481
ϕ_4	1.0000	1.0000	—.8640	.3847

third mode and frequency from a matrix B_3 similarly obtained from B_2, etc.

This process is used in the following to obtain the second and fourth modes and frequencies, and the remaining frequency without its mode, in the present example. B_1 is the square matrix on Page 162, Part I, which is here repeated:

$$\begin{bmatrix} 1.2 & .9 & .7 & .6 \\ 1.2 & 1.9 & 1.7 & 1.6 \\ 1.2 & 1.9 & 2.7 & 2.6 \\ 1.2 & 1.9 & 2.7 & 3.6 \end{bmatrix} \times 10^{-5} = B_1$$

Iteration by premultiplication on B_1 is performed by first selecting any arbitrary row of numbers and applying the procedure described on Page 161 of Part I. For example, selecting the arbitrary row [.5 .7 .9 1.0], the first premultiplication is as follows:

$$[.5 \quad .7 \quad .9 \quad 1.0]\begin{bmatrix} 1.2 & .9 & .7 & .6 \\ 1.2 & 1.9 & 1.7 & 1.6 \\ 1.2 & 1.9 & 2.7 & 2.6 \\ 1.2 & 1.9 & 2.7 & 3.6 \end{bmatrix}$$

$$= [3.72 \quad 5.39 \quad 6.67 \quad 7.36]$$

Normalizing the result yields a new row

$$\left[\frac{3.72}{7.36} \quad \frac{5.39}{7.36} \quad \frac{6.67}{7.36} \quad \frac{7.36}{7.36}\right] = [.506 \ .733 \ .907 \ 1.000]$$

Repeating the premultiplication until the values converge so that alternate columns show negligible differences, the following row matrix results

$$[3.78304 \quad 5.48164 \quad 6.74913 \quad 7.44388]$$

Note that the natural frequency for the first mode is given by the relation $10^5/\omega_1{}^2 = 7.44388$ (compare with the solution on Page 162, Part I). The figures in the row, however, tell nothing else about the first mode.

Normalizing yields the row matrix

$$[.50821 \quad .73640 \quad .90801 \quad 1.00000]$$

Thus, $I - E_1$ is

$$\begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ -.5082 & -.7364 & -.9080 & 0 \end{bmatrix} = I - E_1$$

and $B_2 = B_1\ (I - E_1)$ is found by multiplying, row by column, giving the result

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$$\begin{bmatrix} .89508 & .45816 & .15520 & 0 \\ .38688 & .72176 & .24720 & 0 \\ -.12132 & -.01464 & .33920 & 0 \\ -.62952 & -.75104 & -.56880 & 0 \end{bmatrix} \times 10^{-5} = B_2$$

Iteration by column postmultiplication on B_2 yields, in the final two columns:

−1.33224	−1.10700
− .96577	− .80249
.20338	.16899
1.20347	1.00000

This gives $\omega_2^2 = 10^5/1.20347 = 83093$, $\omega_2 = 288.26$ radians per second, and $f_2 = 2753$ cycles per minute, again agreeing with previous results. The final column which gives the mode is for practical purposes the same as that in TABLE III.

Iteration by row premultiplication on B_2 yields the final row

$$[1 \quad .93760 \quad .44781 \quad 0]$$

The normalizing this time is done by dividing each time by the *first* terms. Thus E_2 is formed by placing the normalized row in the first row, and $I-E_2$ becomes

$$\begin{bmatrix} 0 & -.93760 & -.44781 & 0 \\ 0 & 1 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} = I - E_2$$

Then $B_3 = B_2(I-E_2)$ is

$$\begin{bmatrix} 0 & -.38107 & -.24563 & 0 \\ 0 & .35902 & .07395 & 0 \\ 0 & .09911 & .39353 & 0 \\ 0 & -.16080 & -.28689 & 0 \end{bmatrix} \times 10^{-5} = B_3$$

Iteration by column postmultiplication would yield the third mode which, though not calculated here, is shown by figures in TABLE IV. If only the frequency is desired, iteration by row premultiplication on B_3 gives the final rows

$$[0 \quad .43949 \quad .46364 \quad 0]$$
$$[0 \quad .94791 \quad 1 \quad 0]$$

Here the normalizing is done by dividing each time by the *third* term, so that $10^5/\omega_3^2 = .46364$, $\omega_3 = 464.42$, and $f_3 = 4435$ cycles per minute. Also $I-E_3$ is formed after placing the normalized row in the *third* row:

$$\begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \\ 0 & -.94791 & 0 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} = I - E_3$$

The final matrix, $B_4 = B_3\ (I-E_3)$, contains only one nonzero column:

$$\begin{bmatrix} 0 & -.14823 & 0 & 0 \\ 0 & .28892 & 0 & 0 \\ 0 & -.27392 & 0 & 0 \\ 0 & .11115 & 0 & 0 \end{bmatrix} \times 10^{-5} = B_4$$

Iteration on this is unnecessary, the immediate result being $\omega_4^2 = 10^5/.28892 = 346116.5$, $\omega_4 = 588.32$, and $f_4 = 5618$ cycles per minute. This follows because the matrix B_4 would occur in an equation of the form of Equation 8*b* (Part 1). In the nonmatrix form (Equation 8*a*) the second equation would be

$$\phi_2 = .28892\ \phi_2 \times \omega_4^2 \times 10^{-5}$$

To find the mode, it may be noted that $\phi_1 = -.14823/.28892 = -.5130$, $\phi_2 = 1.0000$, $\phi_3 = -.27392/.28892 = -.9481$, and $\phi_4 = .11115/.28892 = .3847$. Substitution of these values in Equation 5 gives $\phi_0 = .0127$. The mode is indicated in TABLE IV.

In closing it may be noted that the same techniques discussed in the foregoing will yield the critical shaft frequencies in transverse vibration. It is only necessary to replace the torsional influence coefficients $\theta(i, j)$ by an analogous table of *deflection influence coefficients* $\delta(i, j)$ which represent the linear deflection at i due to a unit transverse load at j. Then the moments of inertia of the disks are replaced by their masses W/g, and their rotational coordinates ϕ_n, by linear deflection coordinates y_n. Assuming a fixed crankcase or support, no coordinate y_0, corresponding to ϕ_0, need be considered. The orthogonality condition, if needed, is $\Sigma(W/g)\, y\, y' = 0$. The remainder of the technique then holds without change.

Actual derivation of the $\delta(i, j)$ requires, in general, solution of a statically indeterminate beam system having several supports. This subject is not within the scope of this Data Sheet.

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SHAFT SERRATIONS: The Engineering Data sheet on "Standard Dimensions for Straight Shaft Serrations" (M.D., Nov., 1944) was based on the standards published in the 1943 S.A.E. Handbook. Since that time the S.A.E. standards have been revised, hence the dimensions in the Data Sheet no longer are in accord with the standard. It is suggested that readers desiring to follow the standard, check the dimensions now given in the 1945 S.A.E. Handbook.

MACHINE DESIGN'S
MATERIALS WORK SHEET

FILING NUMBER
12.02

Aluminum Bronzes

ASTM Nos. B148-44T, B150-44T and B169-44T

AVAILABLE IN:

(B148-44T) .... Sand castings
(B150-44T) .... Rods, bars and shapes
(B169-44T) .... Sheet and strip

ANALYSES

SAND CASTINGS (B148-44T)

	Cu	Al	Fe	Ni	Si	Mn	Sn	Impurities	Total of Named Elements
Type 9A	86 min	8.5-9.5	2.5-4.0	....	....	....	....	....	99 min
Type 9B	86 min	9.0-11.0	.75-1.5	....	....	....	....	....	99 min
Type 9C	83 min	10-11.5	3-5	2.5 max	....	.5 max	....	....	99.5 min
Type 9D	78 min	10-11.5	3-5	3-5.5	....	3.5 max	....	....	99.5 min

Note: Types 9B, 9C and 9D respond to heat treatment.

RODS, BARS AND SHAPES (B150-44T)

	Cu	Al	Fe	Ni	Si	Mn	Sn	Impurities
	78-93	6.5-11	4 max	5.5 max*	2.25 max*	2 max	.6 max	.5 max

SHEET AND STRIP (B169-44T)

	Cu	Al	Fe	Ni	Si	Mn	Sn	Impurities
Alloy A	92-96	4-7	.5 max	....	....	....	....	.5 max
Alloy C	90-93	7-9	.5 max	....	....	....	....	.5 max

*When both silicon and nickel are present in this alloy, only one shall be in excess of .25 per cent.

PROPERTIES

TENSILE STRENGTH

(*minimum, psi*)

RODS AND BARS		
Type I, .5-in. and under diam or thick		80,000
Over .5 to 1-in. diam or thick		75,000
Over 1-in. diam or thick		72,000
Type II, (Rounds only) .5 to 1-in. diam		100,000
Over 1 to 2-in. diam		90,000
Over 2 to 4-in. diam		85,000
SHAPES		
All sizes (Type I)		75,000
SHEET AND STRIP		
Alloy A, .5-in. and less thick, 30-in. and less wide (hard)		60,000
.5-in. and less thick, over 30-in. wide (hard)		55,000
Over .5-in. thick, all widths (hard)		50,000
All thicknesses and widths (soft)		45,000
Alloy C, .5-in. and less thick, 30-in. and less wide (hard)		65,000
.5-in. and less thick, over 30-in. wide (hard)		60,000
Over .5-in. thick, all widths (hard)		55,000
All thicknesses and widths (soft)		50,000
SAND CASTINGS	**As Cast**	**Heat Treated**
Type 9A	65,000	....
Type 9B	65,000	(9B-HT) 80,000
Type 9C	75,000	(9C-HT) 90,000
Type 9D	90,000	(9D-HT) 110,000

MACHINE DESIGN is pleased to acknowledge the collaboration of the following companies in this presentation: The Ajax Metal Co.; The American Brass Co.; Ampco Metal, Inc.; Bridgeport Brass Co.

YIELD STRENGTH

(*minimum, .5% elongation under load, psi*)

RODS AND BARS	
Type I, .5-in. and under diam or thick	40,000
Over .5 to 1-in. diam or thick	37,500
Over 1-in. diam or thick	35,000
Type II, (Rounds only) .5 to 1-in. diam	50,000
Over 1 to 2-in. diam	45,000
Over 2 to 4-in. diam	42,500
SHAPES,	
All sizes (Type I)	30,000
SHEET AND STRIP	
Alloy A, .5-in. and less thick, 30-in. and less wide (hard)	24,000
.5-in. and less thick, over 30-in. wide (hard)	22,000
Over .5-in. thick, all widths (hard)	20,000
All thicknesses and widths (soft)	17,000
Alloy C, .5-in. and less thick, 30-in. and less wide (hard)	27,000
.5-in. and less thick, over 30-in. wide (hard)	25,000
Over .5-in. thick, all widths (hard)	22,000
All thicknesses and widths (soft)	20,000

SAND CASTINGS	As Cast		Heat Treated
Type 9A	25,000		.....
Type 9B	25,000	(9B-HT)	40,000
Type 9C	30,000	(9C-HT)	45,000
Type 9D	40,000	(9D-HT)	60,000

ELONGATION IN 2 INCHES

(*minimum, per cent*)

RODS AND BARS	
Type I, .5-in. and under diam or thick	15
Over .5 to 1-in. diam or thick	15
Over 1-in. diam or thick	20
Type II, (Rounds only) .5 to 1-in. diam	10
Over 1 to 2-in. diam	12
Over 2 to 4-in. diam	15
SHAPES	
All sizes (Type I)	15
SHEET AND STRIP	
Alloy A, .5-in. and less thick, 30-in. and less wide (hard)	25
.5-in. and less thick, over 30-in. wide (hard)	25
Over .5-in. thick, all widths (hard)	30
All thicknesses and widths (soft)	40
Alloy C, .5-in. and less thick, 30-in. and less wide (hard)	20
.5-in. and less thick, over 30-in. wide (hard)	20
Over .5-in. thick, all widths (hard)	25
All thicknesses and widths (soft)	30

SAND CASTINGS	As Cast		Heat Treated
Type 9A	20		..
Type 9B	20	(9B-HT)	12
Type 9C	12	(9C-HT)	6
Type 9D	6	(9D-HT)	5

CHARACTERISTICS

In wrought form, aluminum bronzes are among the most important copper-base alloys furnished to the aircraft industry. They possess strength and ductility similar to medium carbon steel and, in addition, offer high resistance to the corrosive action of the atmosphere, salt water, sulphuric acid, and other chemicals. Their resistance to scaling or oxidation at elevated temperatures is excellent—better in fact than that of any other copper-base alloy—and this resistance increases with aluminum content. They are readily forged and hot rolled. Some can be cold rolled and some are susceptible to hardening by heat treatment. They have good bearing qualities, hardness, and resistance to shock and fatigue. Weights of aluminum bronzes range from five to ten per cent less than other common brasses and bronzes. The standard alloy covered by ASTM Spec. No. B150-44T is essentially a hot-working material used for making strong hot forgings. It is very hard and does n[ot] lend itself to severe cold working.

Type 9A casting alloy in the as-cast condition has excellent resistance to corrosion by acids such as tannic, sulphurous, vegetable and fruit. Types 9B, 9C and 9D when heat treated have improved tensile strength but ductility i[s] lowered. The color of aluminum bronzes may be likened to that of 10-carat gold and they will take beautiful oxidized finishes.

APPLICATIONS

These alloys are excellent for applications requiring hig[h] tensile properties plus good corrosion resistance. Typica[l] parts advantageously made of aluminum bronze are: Valv[e] stems and guides, propeller-blade bolts, air-pump part[s], condenser bolts, slide liners, bushings, propeller-hub cones, spark plug inserts and nuts. The casting alloys make goo[d]

PHYSICAL CONSTANTS

(nominal values)

	Sand Castings (as cast)		Rods, Bars, Shapes	Sheet and Strip	
	Type 9A	Type 9B		Alloy A	Alloy C
elting Point (deg F)	....	....	1840-1930	1940	1904
pecific Gravity	7.3-7.5	7.3-7.65	7.58-7.78	8.17	7.78
ensity (lb per cu in.)	.264-.271	.264-.276	.274-.281	.295	.281
hermal Conductivity					
(Btu/sq ft/sec/deg F/in) at 68 F	.089-.104	.104-.112	.073-.137	.153	.137
(cal/sq cm/sec/deg C/cm) at 20 C	.11-.13	.13-.14	.091-.17	.19	.17
hermal Coef. of Linear Expansion					
per deg C	.000017 (1)	.0000164-176 (3)	.000017-18 (5)	.0000179 (5)	.0000179 (5)
per deg F	.0000095 (2)	.0000091-98 (4)	.0000094-100 (6)	.0000099 (6)	.0000099 (6)
lectrical Resistivity at 20 deg C (ohms per mil ft)	74-86	69-86	70-138	61	70
lectrical Conductivity at 20 deg C (% of Int'l Ann'ld Copper Std)	12-14	12-15	7.5-14.8	17	14.8
odulus of Elasticity (psi, millions)	16-18	14-16	....	....	....

(1) 21 to 121 C (2) 70 to 250 F (3) 21 to 260 C
(4) 70 to 500 F (5) 25 to 300 C (6) 77 to 572 F

pur, helical, bevel and internal gears, especially where ated with hardened steel gears. In steel mills these alloys e used for stripper nuts, slippers and heavy-duty feed uts.

FABRICATION

ACHINABILITY:†

Alpha aluminum bronzes (containing less than about 7.5 er cent aluminum), produce tough continuous turnings imilar to those of the alpha-tin bronzes, and are best achined with a rake angle of 12 to 15 degrees, as used or steel. However, machining operations more commonly re applied to alloys containing more than 7.5 per cent luminum. With these alloys, cutting is accompanied by ransverse shearing across the turnings which renders them rittle, often breaking them up into short chips. This shearing induces some vibration in the tool, the effects of which re aggravated when excessive rake angles are employed, articularly if the setup of the tool and work is not sufficiently rigid. For such materials, a top rake of about 8 egrees gives the best all-around results. This angle may, ith advantage, be increased for light finishing cuts or even or roughing cuts if the setup is rigid. On the other hand t may have to be reduced for heavy cuts on the hardest lloys.

Machining properties of aluminum bronzes are somewhat superior to those of ferrous materials having equivalent mechanical properties. Nevertheless, tool wear is more apid and heating is more pronounced than with milder rasses and bronzes. Hence, for production work, high-duty materials such as tungsten high-speed steel, cobalt igh-speed steel or tungsten-carbide are recommended. ungsten-carbide tools are particularly suitable for finishing or roughing cuts on bar stock but are not recommended or roughing irregular surfaces unless the setup is extremely igid. Use of a soluble oil or even a straight machine oil ombining good cooling and lubricating properties, will be ound to increase tool life considerably.

†*Aluminum Bronze*, Copper Development Association, London, W.C. 2.

DRILLING:

When properly sharpened, standard high-speed steel drills can be used successfully on aluminum bronzes. Drills are ground to zero rake angle except in the case of drilling extremely hard grades, when the angle should be minus three to minus five degrees. The same sharpening practice is applied to tungsten-carbide-tipped drills. In drilling deep holes the drill is cleared at frequent intervals, otherwise chips pile up in the drill which may result in breakage. However, when drilling holes ½-inch or less deep it is not necessary to clear the drill. In view of the expansion which occurs on heating and the possibility of the drill binding in the hole, drills sometimes are ground slightly off center to provide additional clearance. Using high-speed steel drills, aluminum bronze can be drilled from 70 to 150 surface feet per minute. Using tungsten-carbide-tipped drills, this can be increased by about 40 per cent. Sulphur-free coolants are used to advantage in all drilling operations.

REAMING:

It is important to leave enough stock for reaming, because insufficient stock will create a burnishing or rubbing action that generates heat and results in undersize holes. Reamers should be provided with more back taper than is usual for steel so as to reduce the tendency toward binding. Amount of total stock to be left for reaming should be from .012 to .018-inch. Feeds vary from .005-inch per revolution for ¼-inch diameter holes to about .015-inch per revolution for 1-inch holes. For high-speed steel reamers a cutting speed of 50 surface feet per minute is suitable. This may be increased to from 80 to 175 feet per minute for tungsten-carbide-tipped reamers.

TAPPING:

Proper tap design is most important in tapping aluminum bronzes. Taps with twelve to eighteen-degree spiral points extending beyond the first full thread should be used at all times. They should have a rake angle of 3 to 5 degrees for the softer bronzes, but for the harder grades, zero rake angle is recommended with a 10 to 15-degree chamfer for a length of two to three threads. In tapping fine to mod-

erate threads, speeds of 30 to 80 surface feet per minute are recommended. When tapping coarse threads, this speed should be reduced by about 25 per cent. Sulphur-free coolants containing good lubricating properties should be generously applied.

STAMPING AND PERFORATING:

In the low-aluminum, single phase, annealed state, it is possible to perforate 1/32-inch thick aluminum bronze sheet with holes whose minimum diameter is equal to the thickness of the sheet. Die and punch clearances are the same as for mild-steel work. For general-purpose stamping, anything that can be stamped from mild steel also can be stamped from aluminum bronze providing it is in the annealed state described above. Standard die and die-block clearances used on steel are acceptable for aluminum bronze. The stamped edge will be less distorted or "necked down" than generally is the case with steel. Usually heavier stamping presses are required than for mild steel.

BENDING, FORMING AND SHEARING:

For welded fabricated structure work, aluminum bronzes with low aluminum content readily lend themselves to cold bending, forming, punching, etc. When working plate ⅜-inch thick and heavier, it is of advantage to heat the material above 700 degrees Fahr. in order to save time and utilize smaller bending, rolling and punching equipment. It should be remembered that although preheating is desirable, it is not essential. When shearing ⅜-inch and heavier plate, it is best to use a shear the capacity of which is ⅛-inch greater than one used for mild steel.

DEEP DRAWING:

Alloys A and C of ASTM Spec. No. B169 can be deep drawn, the procedure resembling that used for phosphor bronze. In other words, while the material can be deep drawn, it is inferior in this respect to the usual grades of deep-drawing brass. Material for deep drawing should be specified in soft temper. In this condition deep drawing is readily accomplished, but work-hardening takes place rapidly. Therefore the amount of drawing permissible in a single operation is limited and must be followed by reannealing of the stock, usually after each operation.

SPINNING:

Aluminum bronze can be spun readily when the action is not too severe. On the thinner gages where the radius is not less than 5 to 10 times the thickness of the material, the operation can be performed without intermediate annealing. On heavier material it will require 2 to 5 annealings to accomplish the same result. Except in unusual cases where only meager spinning is required, it is impractical to spin aluminum bronze in thicknesses greater than ⅛-inch.

FORGING:‡

Aluminum bronze is extremely malleable at temperatures between 500 and 850 degrees Cent. The operations of forging or rolling are, moreover, greatly facilitated by the complete absence of oxidation at these temperatures. For hot-working operations, the metal should be heated slowly and evenly and therefore the use of a muffle furnace is recommended. As the alloys suffer no deterioration through being kept hot for long periods, several forgings can be worked at the same time and each piece kept uniformly heated throughout its mass. The best temperature for breaking down lies between 750 and 850 degrees Cent. If heated above 900 degrees Cent. the metal deteriorates permanently and no form of subsequent heat treatment will restore its properties. Forging or rolling below 700 degrees Cent. produces an increase in the ultimate tensile strength with a corresponding reduction in elongation. The metal work-hardens if forged below 700 degrees Cent. If local heat is required, other parts of the forging may be quenched in water. The whole forging should then be stress-relieved at a temperature not lower than 700 degrees Cent. The general rule, true as regards many metals, that wrought material is better than cast, cannot be applied to the aluminum bronzes. Characteristics of any given section of an aluminum bronze part are not improved by hot working when compared with those of a chill casting.

WELDING:

The satisfactory welding of aluminum bronze constitutes one of the most difficult operations in welding practice, primarily because of the refractory oxide films which are formed on the surface of the hot metal. In fact, until recently the welding of aluminum bronze was not generally considered a commercial proposition. Much depends on the flux employed. Various aluminum bronzes can be welded with either the metallic-arc or carbon-arc process when depositing filler metal of similar chemical analysis. When welding aluminum bronze containing 11 per cent or more aluminum, best results are obtained when the base metal is preheated to 300 to 500 degrees Fahr. Aluminum bronze containing less than 11 per cent aluminum does not necessarily require a preheat.

Aluminum bronze weld metal should be deposited with direct current, reverse polarity with metallic arc, and direct current, straight polarity, with the carbon arc.

SOLDERING:†

Soldering of copper alloys containing large amounts of aluminum generally is considered to be difficult. Success often hinges on the amount of aluminum contained. For example, while alloys containing 5 per cent aluminum may be soldered with fair success, those of 8 per cent aluminum content are much more difficult. As is the case with welding, much depends on the flux employed. The benefits attending the use of fluxes can partly be attributed to the blanketing action they offer which prevents the formation of aluminum oxide films in the heating-up stage. It appears that fluxes of a sufficiently low melting point, offering good covering capacity, should behave most satisfactorily. An alternative method sometimes practiced is to plate the aluminum bronze parts with copper and then solder in the normal manner. According to C. H. Meigh the soldering of aluminum bronze can be performed more easily with a solder having a composition of 80 per cent lead and 20 per cent cadmium using conventional fluxes.

BRAZING:†

The above remarks concerning soldering and the advisability of using a flux for protection during heating-up apply also to brazing processes. If normal care is exercised it would appear that there is little difficulty in securing good flow of brazing solders or silver solders onto the base metal.

‡*The Practical Application of Aluminum Bronze*, 1941, McGraw-Hill Publishing Co. Ltd., London, W.C. 2.

HEAT TREATMENT

The casting alloys respond to heat treatment in much the same manner, but to a lesser extent, as the fabricated alloys. However, heat treatment has an especially favorable effect on yield properties, and where yield strength is of special importance in castings made from the 88/9/3 alloy (ASTM Spec. No. B148-44T, Type 9A), for example, there is an advantage in quenching from 850 to 900 degrees Cent. followed by reheating at about 600 degrees Cent. By this means the yield strength may be increased from a typical 11 to 14 tons per square inch to about 18 tons per square inch with little change in tensile strength and elongation.

Heat treating of aluminum bronze generally consists of a water quench from 1500 to 1700 degrees Fahr. followed by a "draw" at temperatures of from 1000 to 1200 degrees Fahr. This treatment results in higher tensile strengths, higher yield strengths, and lower percentages of elongation with approximately a 20-point increase in brinell hardness as compared to the "as cast" or "as annealed" physical values. The fact should not be overlooked that with large castings it may not be possible to carry out heat treatment, as the necessary equipment for heating and quenching often is not available. This naturally imposes limitations on the application of heat treatment to some aluminum-bronze castings.

ANNEALING AND PICKLING

Annealing of aluminum bronze usually is employed to "balance" the phase (and physical properties) of castings of varying cross sections as well as to accomplish stress relief. Annealing of cold-worked alpha aluminum bronzes may be conducted in any conventional type of furnace capable of operating at the necessary temperature of 600 degrees Cent. In heating to this temperature there is practically no oxidation of the metal owing to the protection afforded by the aluminum oxide films formed in the early stages of heating. However, carbonaceous deposits often mar the bright appearance of annealed products, and it is particularly recommended that precautions be taken to remove oil lubricants and also that the annealing not be performed in open-flame furnaces in which carbonaceous deposits may be formed on the metal. If deep-drawing processes have to be performed, bright annealing is advisable in order to avoid possible scoring difficulties.

When annealed under clean conditions aluminum bronze does not require pickling, at least in the manner of other copper alloys. It is, however, conventional to pickle in warm 5 per cent sulphuric acid, although such acids obviously are not capable of reacting with aluminum oxide. With "dirty" annealed aluminum bronze, pickling often is carried out in acid solutions containing potassium or sodium bichromate; such solutions etch the products rather deeply and the "dirty" films are afterwards removed by wiping. In other instances abrasive methods are employed for removal of undesirable surface layers, since all pickling methods suffer from the disadvantage that the exposed portions of clean metal are attacked at a more rapid rate than the areas protected with oxide films. Physical properties of the metal "as annealed" generally agree closely with those of the metal in the "as cast" state.

RESISTANCE TO CORROSION

Aluminum bronzes offer better corrosion resistance than pure copper to many corrosive media. They show excellent resistance to industrial and marine atmospheres, sea water and fresh waters. They offer good resistance to corrosion by most acids, salts and alkalies and are useful in handling many organic compounds including alcohol, phenol, cresol, fatty acids and organic salts. Like other copper alloys they are not suitable for handling ammonia, nitric acid, chromic acid, acid chromates, ferric salts, or mercury salts.

GALVANIC CORROSION

Aluminum bronzes will exhibit galvanic corrosion effects similar to pure copper and in general may be safely coupled with copper and other copper alloys. When large areas of aluminum bronzes are connected galvanically to iron, zinc, aluminum or high brass in corrosive environments, the whole assembly should be painted or some other method of insulating the aluminum bronze from the other metal should be employed to protect the other metal from rapid galvanic attack.

MATERIAL DESIGNATION[§]

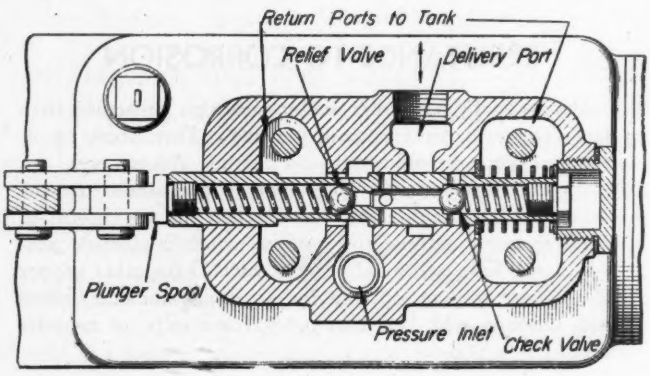
ASTM No.	SAE	AMS	Army-Navy Aero. Board	U. S. Air Forces	U. S. Navy	U. S. Army	Federal
B148-44T							
Type 9A	68, Gr A	....	....	....	....	....	....
9B	....	....	....	....	....	....	QQ-B-671a, Cl. B
9C	....	4640	....	....	....	....	....
9D	....	....	AN-QQ-B-672	11076	....	....	....
B150-44T							
Type I	701,Gr.B,Opt.1	....	AN-B-16,Opt.1	....	46B17(INT) Gr.B,Opt.1	....	QQ-B-666,Gr.B Opt. 1
Type II	701,Gr.B,Opt.2	....	AN-B-16,Opt.2	....	46B17(INT) Gr.B,Opt.2	....	QQ-B-666,Gr.B Opt. 2
B169-44T							
Alloy A	701, Gr.A	....	....	....	46B17(INT) Gr.A 46B17b, Gr.A	....	QQ-B-666,Gr.A
Alloy C	....	4631(wrought) 4630A(wrought) 4632 (bars)	....	....	46B17b,Gr.B	....	....

§From *Cross-Index of Chemically Equivalent Specifications and Identification Code*, published by General Motors Corp.

Noteworthy Patents

Valve Combines Essential Controls

A HYDRAULIC control valve which combines all the parts necessary to provide two-way control, check and relief valving within the hollow plunger spool is covered by patent 2,362,945 recently assigned to the Hydraulic Control Engineering Co. As shown in the accompanying illustration, the valve is of the self-centering type, blocking in neutral position the delivery, check



Control, check and relief are featured in this one valve

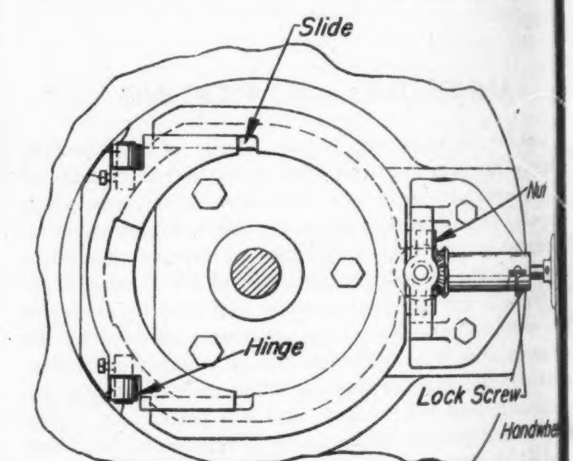
and relief ports while bleeding the pump pressure back to tank. Movement of the spool to the left delivers oil pressure from the pump to a cylinder or ram by unseating the ball check. Failure of delivery pressure for any reason merely reseats the ball check, effectively holding a loaded ram. Movement of the spool to the right allows the oil returning through the delivery port to drain back to tank by again unseating the check. During the operating cycle the relief valve remains closed as long as the preset pressure is not exceeded, but operates to relieve any abnormal pressure shock or surges as they arise. Preset discharge pressure from the relief valve can be utilized to operate some external device in sequence with a primary cylinder or ram if desired.

Roller Cam Is Adjustable

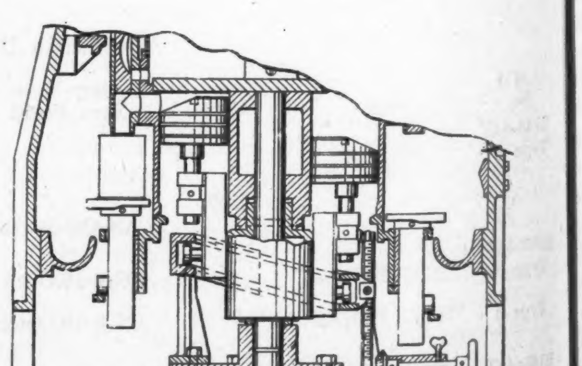
DETAILS of an improved cam structure for container filling machines which may easily be adjusted during production are covered by patent 2,307,214 assigned to the Food Machinery Corp. The cam is accurately adjustable for varying the amount of fill required without causing any appreciable loss in operating time.

Designed with a channel-shaped cross section to guide filling piston drive rollers, the substantially circular cam assembly shown in the accompanying illustration, is made in two sections with closely fitted slides mating to form an unbroken track surface. One half of the cam sect is hinged to a permanent support while the mating h is pinned to a nut which can be adjusted vertically a hand-operated screw. As the nut is raised or lowe by the screw, the cam assembly in turn pivots about hinge—contracting or expanding the telescoping sl —and raises or lowers to increase or decrease the t vertical movement of the drive rollers with each re lution of travel about the cam path.

Micrometric variation of the total cam pitch to pro a suitable piston stroke is therefore a simple operati Changes in cam position on the vertical screw effe only a small telescoping action between the two ca sections. Track alignment and continuous roller path maintained throughout the adjustable range.



Plan View of Cam



Built of two telescoping sections, this circular path roll cam may be adjusted without disturbing the continuity the two-piece roller path

PROFESSIONAL VIEWPOINTS

"... tolerances can be held"

To the Editor:

The article, "Specifying Rational Tolerances for Interchangeability, Low Cost" by F. A. Wedberg, in the January issue, was very interesting. However, I believe a number of points are worthy of further consideration.

A machine is capable of producing parts within close tolerances whether dimensioned with angles or, as the author recommends, with offsets. Checking or locating work to a given angle, within close angular tolerances, may be done with the use of a sine bar and sine table. Too, many machines are designed to work directly to angular variations.

Avoidance of the use of steel fixtures for producing large aluminum alloy assemblies, as proposed by the author, is desirable, but unfortunately he offers no suggestion to correct this condition. Without a doubt, it would be ideal to construct jigs and fixtures with a material having the same coefficient of expansion as the part being produced in it, but obviously such a procedure would be more expensive in time and material.

Careful consideration should be exercised in the adoption of a tolerance reserve policy. Retaining a tolerance reserve for inspection and salvage purposes as suggested, might result in a bad psychological effect on plant personnel when it became generally known that acceptability beyond specified limits was approved by the inspection department. Such a system would require close coordination and cooperation between the departments involved, but I agree with the author that after once having been established on a sound basis it would result in much more efficient production.

—W. E. Schaefer, *Production Design Engineer*
The Glenn L. Martin Co.

To the Editor:

Mr. Schaefer raises some very good points upon which I am glad to offer the following comments.

In determining or checking an angle by the use of angular measurements as in the case of a protractor, an accuracy of possibly one-quarter of a degree may be expected and this may be reduced somewhat with instruments of greater precision. The same angle may be established with much greater accuracy by using offset dimensions. For example, if a point is located on a 10-inch radius and the offset is measured within plus or minus .010, the angle will be accurate to about three minutes. Should the 10-inch dimension be 20 inches, the same degree of measuring accuracy will give the angle to about one and one-half minutes.

It is recognized that designers have been handicapp by wartime aluminum alloy shortages and consequent strictions on use of these materials. It may also be so what impracticable and unnecessary to fabricate large tures from aluminum alloy material. However, we ge erally find that on many large airframe assemblies it necessary to hold extreme accuracy on only a relativ few points to avoid adverse temperature effects. A typic example of the necessity for holding a series of dimensio to close limits is that of aileron or wing-flap hinges. In case of this kind, long aluminum alloy rods or rectangul sections have been used on a steel fixture by anchori one end and permitting the other end to float.

The principle of retaining a tolerance reserve is open some controversy, but its use is based on a combination psychology and provision against unforeseen or unpredic able errors. Drawing tolerances customarily represent desirable result. The tooling and manufacturing orga zations base their equipment and methods upon obtaini that result. However, it has been found quite desiral to retain a reserve tolerance to cover: (a) Early produ tion deviations which can be later eliminated by refin ment of tools and processes, and (b) deflections of t type which will occur due to riveting, welding, or he treating strains and which can never be entirely eli inated. A case in point is that of an all-metal fusela whose wing or landing gear attaching points are held ground pins during fabrication. Upon completion of t assembly, there will invariably be some deflection and co sequent misalignment of any fitting upon withdrawing t pin. A reserve tolerance of .015 or .030 or even mo may be allowed for this misalignment, recognizing th some slight springing will be required to assemble t unit to its mating part. Reserve tolerances also cover: (Deviations in an assembly like a long aileron or flap, whi although built in a fixture that holds all hinge points line, will deflect to a certain degree due to fabricati strains as well as its own weight when removed from th fixture; (d) allowance on hinge alignment that may used by inspection for acceptance or to indicate the d gree to which a hinge may be sprung during final assemb operations without imposing unreasonable strain. Ine perience of personnel, occasioned by the tremendous wa time expansion, has brought on the need for more co plete written information of this type, approved by th proper engineering and customer personnel. An outgrowt of this need is the Army Air Force's Materials Review Pro cedure which definitely provides for properly approved written reserve tolerances.

—F. A. Wedberg, *Tech. Ass't. to Dir. of Eng.*
Curtiss-Wright Cor

Magnesium alloys are easily joined by every common method

Riveting—gas, arc, spot and flash welding all do the job

Matchless lightness has earned for magnesium an industry-wide reputation. It's when you come to build this lightness into your product that you first fully appreciate magnesium's many important fabrication advantages.

Easy joining is a major one. Magnesium readily lends itself to every joining method in common use, including riveting and gas, arc, spot, and flash welding. Procedures are very similar to those employed with other metals.

Riveting is the method most widely used for joining magnesium sheet and extrusions, and various Dowmetal Magnesium Alloys in these forms—as well as sand castings—can also be gas and arc welded. Spot and flash welding each serve definite fabrication requirements.

Dow has taken active part in the development of these techniques, and the resultant data is now available to you. The nearest Dow office will give you technical assistance in the best procedures to use in your own product.

DOWMETAL

magnesium

THE METAL OF MOTION

AGNESIUM DIVISION, THE DOW CHEMICAL COMPANY, MIDLAND, MICHIGAN

York • Boston • Philadelphia • Washington • Cleveland • Detroit • Chicago • St. Louis • Houston • San Francisco • Los Angeles • Seattle

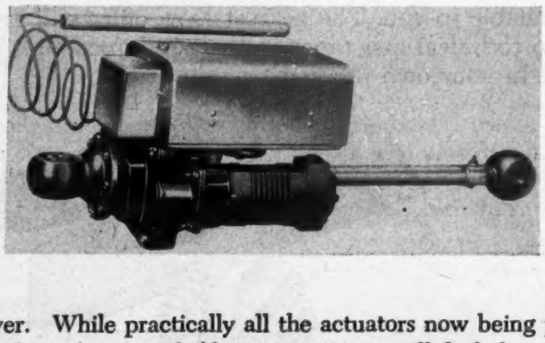
New PARTS AND MATERIALS

Oiltight Pushbuttons

DESIGNED PRIMARILY for group mounting on machinery or control enclosures, a new line of Class 9001 Type T oiltight pushbuttons has been introduced by the Industrial Controller division, Square D Co., 401 North Richards street, Milwaukee 12. While oiltightness is the principal feature of these pushbuttons, there also are other advantages. Type T units are compact and can be mounted on closer centers than previous types. All terminal screws can be reached with a screw driver without going in at an angle. Quick and easy installation is another advantage. The unit is inserted through the panel and prevented from turning by a dowel. After the legend plate is slipped on, a thread ring clamps the unit into position. Since operating mechanism and contact block are separate units, it is possible to obtain a combination to cover a wide range of circuit requirements with a limited stock of three types of operators and four types of contact blocks.

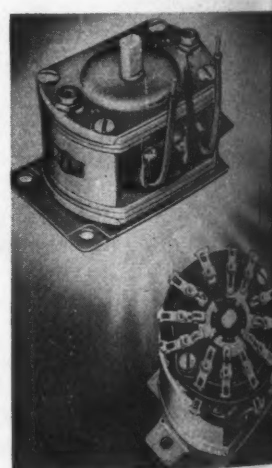
Lightweight Linear Actuators

WEIGHING ONLY 3.05 pounds, Model 400 Series of linear actuators have been built by Lear Inc., Piqua, O., to meet the demand of the aviation industry for a method of converting electrical energy into linear actuating force with least weight and size and the required strength and



power. While practically all the actuators now being produced are for aircraft, design engineers will find them applicable in many postwar products. The actuators operate under loads up to 400 pounds of compression or tension. They require low power, 24-28 volts, and have extremely low current drain on the electrical system. In size, t model is less than 5 inches wide, and less than 7 inc long, including the limit switch control box and ther protector. Extension length ranges from 14⅞ inches almost 25 inches. Control boxes are equipped with li switch control of two, three or more positions. Overtr is eliminated by the company's "Fastop" clutch. A mo ment of a few thousandths of an inch disengages the dri disk of the clutch from the driving disk on the arma shaft. Gear reduction ratios are available in various c binations. The motor, designed for intermittent d runs at average speeds of 9000 to 11,000 revolutions minute. For continuous duty, a motor of special des can be furnished. Some of the applications of the act tors include air filter doors, intercooler shutters, carbure air duct shutters, oil cooler shutters, etc.

Compact, Rotary Relay



OPERATING on a rotating balanced principle instead of the conventional method, a new type of relay has been designed by Price Brothers Co., Frederick, Md. Known as the RO-T-RY relay, it is suitable for applications involving vibration, temperature and humidity conditions. The basic unit is a compact driving mechanism, providing 30 degrees of clockwise or countercl wise rotation. When used to operate switch wafers, makes a relay with a variety of contact arrangeme adaptable for spaced wafer switches or switches in se arate compartments. Where switch wafers are not use a special self-contained coil break switch is provide Overall dimensions of the relay are 2½ x 1½ x ¾ inches.

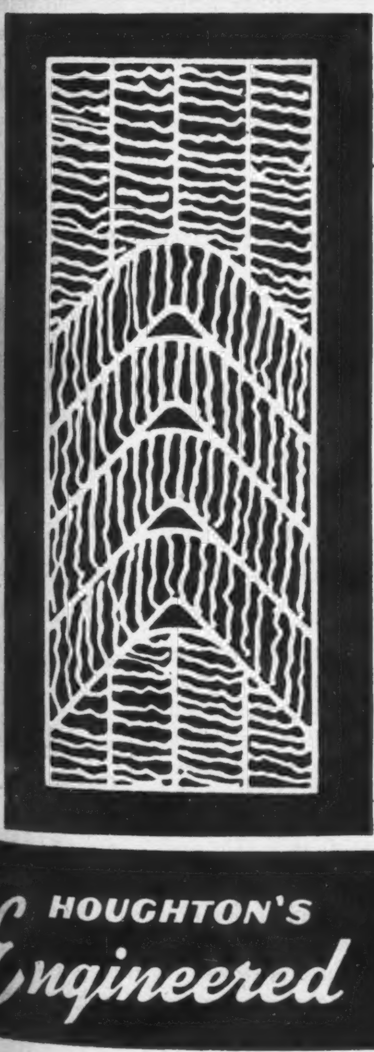
Overload-Release Clutch

AN OVERLOAD-RELEASE clutch and coupling, intr duced recently by The Hilliard Corp., 103 West Fou street, Elmira, N. Y., is based on a new principle, givi instant and complete release when loaded beyond t torques it is adjusted to transmit. In the accompanyi illustration an overload-release coupling is shown, usi

NOW-

LEATHER SUPPORT RINGS FOR "V" PACKING SETS

QUICK DELIVERY • EASILY INSTALLED



In the past it was necessary in assembling "V" Leather Packing installations to have top and bottom metal support rings to hold the packings in place. The shortage of usable metal has too often delayed such installations. Also, in order to install metal support rings, it was necessary to tear down the press, which added to the cost.

Now you can obtain male or female support rings made of laminated plies of leather as shown in the drawing at the left—without delay, and easy to install because the rings can be split.

The leather used in these rings is impregnated to make it impervious to the hydraulic medium. The combination of these support rings with VIM Leather "V" Packings makes a packing installation that will hold at any pressure up to 10,000 PSI.

For full design information, contact the Houghton Man or write us direct. E. F. Houghton & Co., 303 West Lehigh Avenue, Philadelphia 33, Pa. Offices in all principal cities.

HOUGHTON'S Engineered VIM Leather Packings

gear-tooth type, double-engagement, flexible-coupling element. The major parts of the overload mechanism are a hub mounted on the shaft, a sliding jaw ring splined to the hub, a revolving jaw ring which turns on the hub when the jaws are not meshed, a special "dished" actuating spring which applies pressure to the jaws, and an adjusting nut threaded on the hub. Kept in driving contact until the actuating spring starts to deflect in the presence of an overload, the jaws are drawn apart by the special characteristics of the spring after the action of the mechanism has been started. This action, once started, completes itself even though the torque applied to the clutch does not increase. Re-engagement of the mechanism requires lining up of jaws and exerting pressure on the sliding jaw ring to force the jaws together. The spring completes the action, closing the jaws after it has been partially depressed.

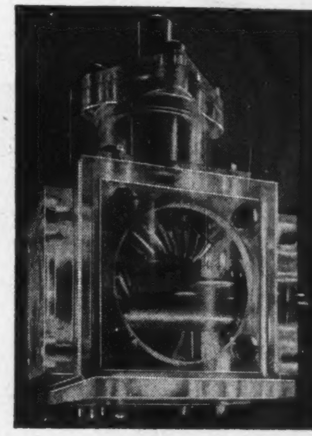
Rubber-Cushion Air Wheel

HAVING LARGE demountable, cushion-type roller bearing rubber tires (of aircraft design), a new air wheel has been introduced by The Rapids-Standard Co. Inc., Grand Rapids, Mich. This AGH wheel, available with a 6-inch diameter and a 2-inch face, has been designed for use on

materials-handling equipment. Equipped with roller bearings and available in axle sizes of ¾, ⅝ and ½-inch with a hub length of 2 3/16 inches, the wheel has a capacity of approximately 250 pounds. Tread of the wheel is held under tension by two locking hub plates of magnesium.

Remote Mechanical Control Units

FOR application on such equipment as cranes, winches, windlasses and steering gears, for opening and closing large motor-operated valves and ventilators and for operation of banks of furnace doors, etc., a standardized system of remotely operated mechanical controls is being offered by M. L. Bayard & Co., 1903 Indiana avenue, Philadelphia 32. The related units—the steady shaft assembly "S" and "P" types, the spiral bevel gear assembly, and the universal shaft assembly—are suitable for power operation where speeds do not exceed 1800 revolutions per minute. Terminal connections are constructed so that rearrangement of the same units can be made without disturbing the assemblies or affecting their internal adjustment. Where required, modifications can be introduced to suit individual requirements. The steady shaft assembly is available in two types: "S" for attaching to surface which is sq[uare] to the shaft, while "P" is for attaching to surface which [is] parallel to the shaft. Shaft ends in both types are [suit]able for mounting of universal shafts or other parts as [re]quired. This assembly can be furnished with differ[ent] shaft diameters and lengths, and with different mount[ing] flanges. The universal shaft assembly is made in leng[ths] ready for installation, or with one end-yoke to be [...]



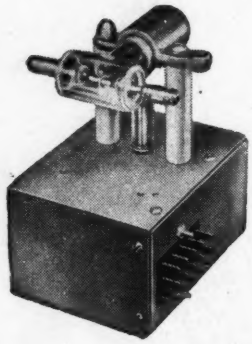
welded after the tube is cut to suitable length. Length[s] arranged to collapse 1 7/16 inches and to extend ¼-in[ch]. Telescoping splined portion between the end yokes p[er]mits ready installation without removing other contr[ol]. Spiral bevel gear assemblies permit full freedom as [...] shaft ends are interchangeable. Either two shaft ends [or] three shaft ends, depending upon the type desired, [...] connected to other control units.

Vertical Discharge Pumps

DESIGNED FOR continuous service under difficult conditions, the vertical discharge type pumps announced recently by Claude B. Schneible Co., 2827 Twenty-fifth street, Detroit 16, are suitable for handling slurries, sludges, abrasive materials and dirty water. Housing is made of abrasion-resistant material, and the top or cover is designed to serve as a strainer. The totally enclosed drive shaft is protected by a quill tube which serves as a structural member and is secured to the mounting plate and housing top. Drive shaft is slotted at the lower end and the one-piece, wear-resistant impeller blade is inserted in the slot and riveted in place. It is connected to the motor with a flexible cou[p]ling located below the mounting plate. Impeller is a[c]cessible and the bottom of pump housing is removable [for] replacement of wearing parts. A venturi type discha[rge] assures constant head pressure. A pipe is connected [to] side discharge and carried upward. Discharge is read[ily] changed to any of four positions. The inverted inlet el[...]

inates gas binding and causes the hydraulic thrust to counterbalance the weight of the revolving parts. Furnished to suit requirements, the motor is splashproof type, with heat-dissipating grids. The pumps are available in sizes from ⅜-inch to 1¼ inches.

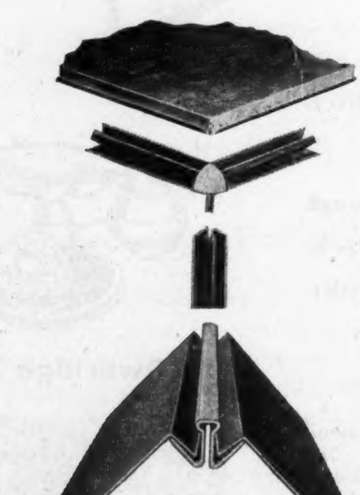
Vacuum-Switch Keying Relay



WHILE THE new Type 78CCA100 vacuum-switch keying relay was originally designed for aircraft use by Struthers-Dunn Inc., 1321 Arch street, Philadelphia 7, it is also applicable to many other fields where units of this type are required. Of simplified and rigid construction, utilizing a minimum of parts, the relay is designed for extreme reliability in holding adjustments. It has seven poles, including one double-throw pole which handles high-voltage, radio-frequency currents by means of a vacuum switch. All high-voltage parts are rounded to reduce corona. According to the company, tests show a life in excess of the minimum five million operations required for units of this type. All parts of the lightweight relay are readily accessible for inspection or adjustment.

Prefabricated Light Assemblies

BASED ON A patented method of snap-assembly which requires no bolts, screws, rivets or welds, a new prefabricated light metal enclosure known as Struc-Lok has been



announced by Lindsay & Lindsay, 222 West Adams street, Chicago 6. This lightweight construction meets the need for applications where light weight and high strength are required. Fabricated in both aluminum and steel, construction consists of three basic parts: Framing, and fittings. The basic principle of the assembly is a lows: Special fittings connect the framing and hold gether while the flanged edges of sheets are snapped frame channels. As edges of sheets snap into place, lock the framing and sheets into position. Struc-L now available with sheets in 26 and 24 gage steel and to .030-inch thickness of 61 ST alloy aluminum. forated or expanded metal sheets may also be used. O ings, louvers, doors and other conventional constru details are easily incorporated, and provision is made using the framing to support shelving, hooks, machine other equipment. A few of the uses of Struc-Lok inc light machinery housings, cabinets for electric and tronic equipment, refrigerators, walk-in coolers, freeze units, furnace casings, kitchen cabinets, air co tioning units, etc.

Miniature Centrifugal Blowers

PARTICULARLY designed for use where a small am of ventilating air is needed to prevent excessive temp tures, a new miniature centrifugal blower, No. 5074 an outgrowth of a military application. Introduced by F. A. Smith Mfg. Co. Inc., P. O. Box 509, Rocheste N. Y., the unit comprises a centrifugal impeller moun

on the shaft of a shaded-pole motor of approximately horsepower. The impeller housing, of pressed steel equipped with a flange for easy mounting. The unit fit within a 4-inch cube. Motor is equipped with oil self-aligning bearings, and the felt-filled reservoir is signed to hold sufficient lubricant for a year's ordin operation. Of shaded-pole two-pole type for continu or intermittent duty, the motor is quiet, making the particularly adaptable for use with electronic assembl

New Bonding Adhesive

ANNOUNCED BY The B. F. Goodrich Co., Akron, a new non-thermoplastic rubber cement, named Plasti 500, is a water and aromatic oil-resistant adhesive bonding metals, wood, plastics and ceramic materials themselves or to each other. Best results are obtain by applying heat with pressure, although heating al will give some degree of adhesion. Bond strength va with the materials being adhered. According to the ma

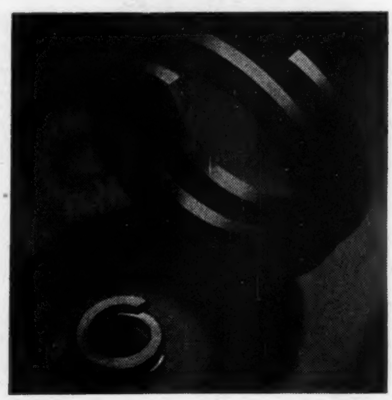
facturers, the new adhesive, used for metal-to-metal bonding, has shown a shear strength of 3250 pounds per square inch; and a tension strength of 4000 pounds per square inch.

Oil and Coolant Strainers

FOR USE ON machine tools for straining cutting oils and coolants and for other installations using flood oiling, a new type of strainer is now available from George Butler Co., 1058 West Washington boulevard, Chicago. No strainer housing is required, the strainer being installed in the tank and the oil or coolant being piped direct from strainer to pump. The strainer units are a combination of wire and cotton, inter-knitted into a mesh. They have a large strainage area, and an exceptional capacity to hold dirt, grit and chips can be provided in the larger units where sufficient space is available for installation. Obtainable in many sizes, the models are rated from two to sixty gallons per minute, and for use with all commercial grades of lubricating oil and coolants.

Double Coil Spring Lock Washers

DOUBLE COIL spring lock washers developed by George K. Garrett Co. Inc., 1421 Chestnut street, Philadelphia 2, offer good reactive spring pressure, plus good resistance to shock, vibration and severe service. In addition to regular uses, they are particularly recommended on grading, bulldozing, agricultural equipment and all



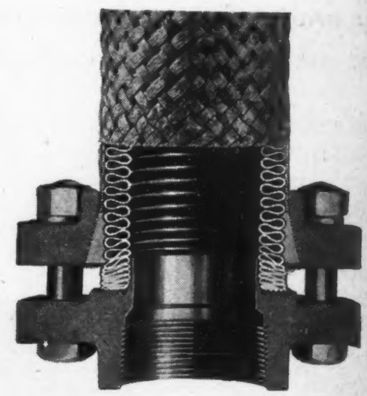
other types of heavy machinery. These double coil washers are furnished in sizes for No. 4 screws up to 1 inch and larger bolts, in any desired finish. Each washer is "torture-tested", that is, subjected to more severe tests than encountered in actual service. The company has also designed and manufactured double coil washers of light sections for many special uses in the electrical and allied industries.

Flame-Resistant Chromate Gasket

CONSISTING OF a felt base, impregnated with a chromate pigmented compound which renders the material flame, fire and corrosion-resistant, a new type gasket has been announced by The Sherwin-Williams Co., 101 Prospect avenue, Cleveland 1. Originally intended as a stitute for low-pressure rubber gaskets in marine ven ing systems, it has since demonstrated its usefulnes many other applications such as joint seals in water, oil and diesel oil systems, as well as gasketing for air and refrigerator doors. This gasket will maintain air sures up to 25 pounds per square inch at normal tem tures. It is dark green in color and is available in thicknesses—1/8 and 1/16-inch.

Detachable Flange for Hose

CONSTRUCTION OF the new detachable flange helical flexible metal hose, announced by Packless M Products Corp., New Rochelle, N. Y., is such that a tive leakproof installation is assured. It is designed to



mit repeated re-use. Assembly operation is simple quick, and with ordinary shop tools. No brazing is quired. No gasket is employed to connect the flange the hose, making a metal-to-metal seal.

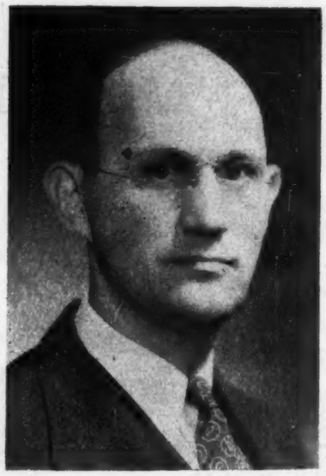
Material for Seals Developed

TO BE USED in seal gaskets, "O" rings, and other t of fuel seals and parts requiring resistance to heat oil, a general purpose stock known as Buna N (14 has been introduced by Los Angeles Standard Ru Inc., 1500 East Gage avenue, Los Angeles. In addi to its high heat and oil-resistant qualities, the materi flexible to 35 degrees below zero.

Silver Plating Aluminum

UTILIZING THE Preplate Process—a developmen the Technical Processes division of Colonial Alloys Philadelphia—silver may be deposited electrolytically aluminum or aluminum alloys, or follow a copper, nic zinc or cadmium deposition. The aluminum is clea passivated, immersed in the Preplate solution for a seconds and then electroplated in the usual manner. plating has good torsion, heat and corrosion resista and adherence. Because of its high rate of conducti silver plating on lightweight aluminums opens up sibilities in the electrical equipment, appliance, trans tation and communications fields.

Thomas C. Leake



Max M. Roensch

Christian E. Grosser

MEN... of... machines

RECENTLY NAMED engineering director of Graham-Paige Motors Corp., Thomas C. Leake has a background of thirty-five years in the transportation field. Since joining Graham-Paige in 1942, he has devoted most of his time to development work on the Navy's LVT (Landing Vehicle Tracked)—the amphibious tractor used in current Pacific landing assaults. Mr. Leake is well qualified for this work, having spent many of his early years as a marine engineer in the Pacific island areas where the LVT is being used, and recent years as a tractor engineer with American firms. He has designed an LVT of his own which has been submitted to the Navy, and is an exponent of "landless transportation", having in mind the use of amphibious tractors as cargo vehicles in Russia and China where there are no bridges or roads. A few years ago he set up such a supply system in China, extending 2000 miles from Chungking to the northern boundary of Outer Mongolia. Before joining the Graham-Paige organization, he had been an engineer with the Eclipse-Pioneer Division of Bendix Aviation Corp., and previously torpedo officer with the British Purchasing Commission.

WIDELY KNOWN in the automotive and petroleum industries as an authority on internal combustion engines, Max M. Roensch has been appointed chief engineer of The Cleveland Graphite Bronze Co. Previous to his new appointment he had been associated with Chrysler Corp. engineering staff for nineteen years. After graduating from Rice Institute, Houston, Texas, with a bachelor of science degree, Mr. Roensch did graduate work in engineering at the U-versity of Michigan, receiving his m-ter's degree in 1926. He then joined Chrysler organization, and remai there until his present appointment. Roensch is a member of the Society Automotive Engineers and the Engin ing Society of Detroit, and is an au of numerous scientific papers which has presented before engineering technical groups. For two years he been vice chairman of the passenger activity of the S. A. E. Detroit Secti and also chairman of committees of S. A. E. War Engineering Board and the Co-ordinating Research Council.

SINCE 1940 consultant on transm sion and general machinery design the Standard Machinery Co., Christi E. Grosser has recently been appoin vice president in charge of engineeri Previously he had been assistant prof sor of mechanical engineering at Mas chusetts Institute of Technology, was connected with Standard in the c

sulting capacity only. He joined the staff of M.I.T. in 1939 as instructor in the divisions of Applied Mechanics and Machine Design and in 1942 became assistant professor of mechanical engineering. Prior to his association with M.I.T., he had been design engineer for Waterbury Tool Co. (1936-1939) on hydraulic transmissions and control systems, principally on Ordnance applications. Before joining Waterbury Tool he had worked on gear transmission problems as design engineer for the Watson-Flagg Machine Co. following his graduation from M.I.T. where he obtained his Bachelor's and Master's degrees in mechanical engineering. In his new post with Standard Machinery Co. he will devote his efforts to hydraulic transmissions and pumps inasmuch as the company for the past five years has given considerable attention to power transmissions for general machinery, developing a number of variable-speed mechanical transmissions, fluid drives and high-pressure pumps.

♦

BYRON CAMPBELL has been transferred by Glenn L. Martin Co., from chief of laboratories, at Omaha, Nebr., to design engineer in its Baltimore plant.

♦

LESLIE D. CALHOUN, design engineer for Busch-Sulzer Bros. Diesel Engine Co., has been appointed assistant chief engineer in charge of design and development, General Machinery Corp.

♦

HECTOR RABEZZANA, who has served for twenty-eight years as chief engineer of AC Spark Plug Division, General Motors Corp., Flint, Mich., has been named president of General Research & Development Co., Fenton, Mich.

♦

OLEG J. DEVORN, previously senior structural engineer of Sikorsky Aircraft, Bridgeport, Conn., is now assistant chief development engineer.

♦

REINHARDT N. SABEE, who had been associated with Micromatic Hone Corp., Detroit, as research engineer, is now chief of research, Special Machine Division, Sav-Way Industries, Centerline, Mich.

♦

FRANK B. FAIRBANKS, president, Horix Mfg. Co., Pittsburgh, has been elected president of the Packaging Machinery Manufacturers institute.

♦

HARRY W. HAHN, one of the leaders in the die-casting field on the Pacific Coast for the past twenty years, has been appointed vice president in charge of engineering and production for the H. L. Harvill Mfg. Co., Vernon, Calif.

♦

FREDERICK J. KNACK, who had been associated with the Fairchild Aircraft Division of Fairchild Engine & Aircraft Corp., has been named vice president in charge of engineering of the Luscombe Airplane Corp., T N. J. Mr. Knack had been vice president and ch gineer for the Luscombe firm for five years prior to when he became connected with the Fairchild A Division.

♦

HENRY B. BRYANS, executive vice president and of the Philadelphia Electric Co., has been re-elected ident of the American Standards association.

♦

J. F. SCHIBLER recently was made assistant chief neer, Taylorcraft Aviation Corp., and will retai former duties as engineering planning supervisor.

♦

JOHN C. STRAUB, who had been associated with th search laboratories division of General Motors Corp. troit for the past thirteen years, has been made re engineer of American Foundry Equipment Co., M waka, Ind.

♦

CLYDE A. PETERSON of Chicago has been name signer in charge of the newly-created radio receiv vision of the Westinghouse Electric Corp. Mr. son has been identified with radio, automobile and trical appliance designing for more than a decade for the last four years has been radio design direc Wells-Gardner & Co.

♦

WILLIAM T. KELLY JR. has been named executive president of the Kellogg Division of American Brake Co. Since graduating from Yale university in 1928 a bachelor of science degree in electrical engineering Kelly has been connected with the organization or its sidiaries.

♦

W. R. DUDA has been appointed vice preside charge of engineering, Continental Foundry & Ma Co.

♦

GORDON LEFEBVRE, president of the Cooper-Bess Corp., has recently been elected to the executive mittee of the Machinery and Allied Products institute.

♦

EVERETT S. LEE, engineer in charge of General E tric's general engineering laboratory at Schenectady, recently re-elected chairman for the Engineers Co for Professional Development for 1945.

♦

W. C. LAWRENCE has been named chief enginee American Export Airlines. Mr. Lawrence formerly been director of development for American Airlines.

♦

S. C. BENNETT has become chief engineer of Lawr Aeronautical Corp., Linden, N. J. He had been ciated with Bell Aircraft at Buffalo and Marietta, since 1942.

Design Abstracts

Development of Silicones

SILICONES—a new class of high polymeric materials —resulted from fundamental research in the field of polymer chemistry bounded by the glasses and silicates on the one hand and by the organic plastics on the other. Interest was stimulated by the development of fibrous glass for electrical insulation, since insulation resins and varnishes of a high order of heat resistance were needed before the maximum advantage could be taken of the thermal stability of fiber glass. It soon became apparent that silicones were natural complements to glass, mica and asbestos in bonding, filling voids and excluding moisture.

One of the first groups of silicone polymers to reach commercial production was the liquid silicones. Several families of these silicone fluids are now available in a wide range of viscosities. They are characterized in general by the properties of low change of viscosity with temperature, low freezing point, unusual inertness and stability in the presence of heat. The silicone fluids are finding application as gage liquids and damping fluids, and in various hydraulic applications. An interesting application is the use of a dilute solution of silicone fluid for rendering ceramic surfaces water-repellent.

Silicone Grease Does Not Melt

A translucent silicone grease of vaseline-like consistency has been developed for use as a lubricant for ignition cables to reduce corona cutting of the insulation and permit easy wiring of ignition harnesses. It is stable to heat and retains its vaseline-like consistency from -40 to 200 degrees Cent. Although it is a soft grease in appearance, it has the unusual property of not melting on exposure to heat. This material also is inert and oxidation resistant. It has no solvent effect upon synthetic insulations or rubber, and tends to prevent the hardening of these materials when heated in contact with air.

Other greases under development are being used for lubricating ball and roller bearings. One type can be used at temperatures as low as -60 degrees Fahr., with high-temperature stability at least as good as the best available organic greases. Another type of silicone lubricating grease is showing stability in ball bearings several times as great as organic greases.

Silicone resins are available in two principal types. The first is the insulating varnish type, which is comparable in physical properties to organic oleo-resinous varnishes. Because of the baking temperatures required to cure the currently available silicone varnishes (200 to 250 degrees Cent.) they must be used with inorganic materials such as glass fiber, asbestos, mica, or ceramics. The silicone varnish is used to coat glass-served magnet wire and fiber-glass cloth, and is used as a binder for building up flexible mica-glass laminated sheet.

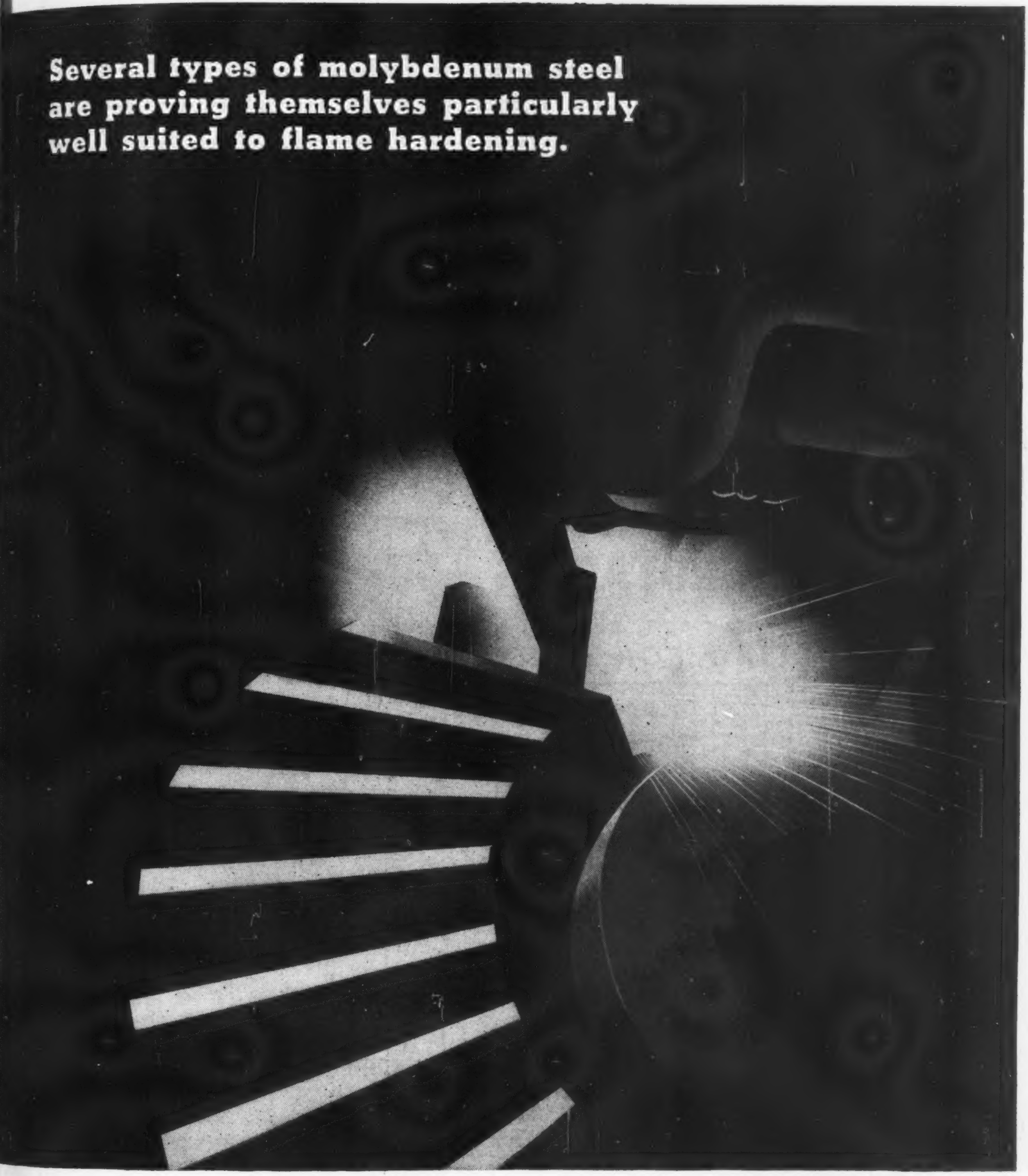
The second type of silicone resin corresponds i eral behavior to the organic thermosetting resins no used to make rigid laminated insulating parts. T more recently developed thermosetting silicone res being used to bind fibrous glass and asbestos lam structures, and to impregnate special coils requiring ness and rigidity.—*From a paper by T. A. Kauppi, Corning Corp., and G. L. Moses, Westinghouse E & Mfg. Co., presented at the A.I.E.E. winter tec meeting in New York.*

Factors Affecting Chafing

CHAFING, galling, or fretting corrosion, as the nomenon is variously called, is a problem whic become more important in aircraft-engine parts a power output has increased and caused a correspo increase in vibration problems. The phenomenon is c acterized by apparent picking out and flow of meta surfaces which are supposedly rigidly clamped on the other. Picking out and flow of metal increases stress concentration in the area where it is present wh of course, leads to reduced strength and, in many c failure. Since the phenomenon has an injurious e on aircraft-engine parts in service operation, an invest tion was initiated to determine conditions which w cause chafing and to discover, if possible, various me of eliminating it or neutralizing its effects.

From these tests and others, the following general c clusions can be drawn:

1. Chafing can be prevented by:
   a. Increasing the compressive load to a value wh all sliding motion is eliminated.
   b. Providing a gasket which can absorb the motion allow motion against steel without pickup.
   c. Providing a coating which can serve as a shear me ber or which can provide an antifriction surfa It is believed that those coatings which are effec in preventing chafing act as an antifriction be ing with many infinitely small balls or rollers.
   d. Providing a plated or otherwise treated surface increase friction and to stop all sliding motion.
2. The severity of chafing:
   a. Increases with increase of motion at a fixed load to the point where the motion applied will cause oil-film wedge action to be maintained.
   b. Increases with increase of compressive load if a g en sliding motion is maintained.
3. Surfaces of unlike metals chafe less than like meta
4. Steel surfaces of the same finish chafe more th surfaces of different finish roughness.—*From a pa by H. C. Gray, Wright Aeronautical Corp., and R. Jenny, Curtiss-Wright Development division, at S.A.E. National Aeronautic meeting in New York.*


Several types of molybdenum steel
are proving themselves particularly
well suited to flame hardening.

BUSINESS AND SALES BRIEF

ESTABLISHMENT of a field engineering office at McGhee avenue, Knoxville 17, Tenn., by representa J. M. Morrison has been announced by Ampco Metal Milwaukee. Mr. Morrison will cover Florida, Georgia, nessee, North and South Carolina.

♦

With headquarters at the home office in Wilmerding, W. V. Walkinshaw has succeeded the late Roland C. J as manager of industrial sales for Westinghouse Air Br Co. Mr. Walkinshaw has been associated with the comp since 1939.

♦

Leroy F. Keely has been appointed general sales man of Howell Electric Motors Co., Howell, Mich. He has more than twenty years experience in the development, and application of electric motors.

♦

Associated with the company for ten years, Leonard has been made a special V-belt representative in the Chic district for Goodyear Tire & Rubber Co. He will serve special representative and sales engineer for original eq ment manufacturers exclusive of farm equipment. Also nounced is the return of Joseph Nieberding from the U. Army. He has replaced Harold Murtaugh as a field rep sentative in Chicago for the mechanical goods division. Murtaugh has been transferred to the St. Louis district wh he will serve as mechanical goods field representative Harrisburg, Ill.

♦

Promotion of W. R. Persons to assistant sales mana has been announced by The Lincoln Electric Co., Clevela In his new position Mr. Persons will assist C. M. Tayl vice president and general sales manager.

♦

Among several organization changes in the New York d trict of General Electric Co. are the following: Horace Zi mer has been named district manager of the industrial di sion, in addition to his present position as district manag of the transportation division. R. B. Ransom has been ma manager of the New Haven office while J. J. Pascher w manage the Hartford office.

♦

Change of name has been announced recently by Westin house Electric & Mfg. Co. Henceforth the company will known as Westinghouse Electric Corp.

♦

The Briggs Clarifier Co., Washington, D. C., has nam the Manning Packing & Supply Co. at 65 South West Se ond avenue, Portland 4, Oreg., to handle distribution Briggs oil clarifiers in Oregon and the southern parts

Washington. Distribution in Wyoming, Colorado, New Mexico and western Nebraska will be handled by the Hendrie & Bolthoff Mfg. & Supply Co., 1635 Seventeenth street, Denver. Oklahoma now will be served by the M. F. Hampton Co., 505 McBirney building, Tulsa 3, Okla. Hoffman Supply Co., P. O. Box 769, Abilene, Tex., has been named to serve northern Texas.

♦

Consolidation of two subsidiaries—Philharmonic Radio Corp. and the Remote Control Division—has been announced by American Type Founders Inc. The Philharmonic corporation will continue its war production of electronic devices. Avery Fisher will remain vice president in charge of sales.

♦

With the Steel Division of the War Production Board in Washington, D. C., since 1942, Donald J. Reese has resumed his duties with the development and research division of The International Nickel Co. Inc. at New York.

♦

Previously chief electrical engineer and supervisor of experimental engineering, John E. Ponkow has been named sales manager by Federal Machine & Welder Co., Warren, O.

♦

According to a recent announcement, Cotner Machine Products Co. of Logansport, Ind., has been acquired by Gerotor May Corp., Baltimore. Md. John C. Cotner, a founder of the Logansport company, has been made a vice president and member of the board of the parent company as well as general manager of the new division. Associated with him in Logansport will be Ruppert Esser, assistant general ager and chief engineer; and Don Thomas, executive engineer. Previously connected with Logansport M Inc., these men have held similar positions during the twenty years.

♦

District sales manager in Chicago since 1921, R. O. has been made central western sales manager for Th liance Electric & Engineering Co., Cleveland. Mr. territory now will include the territory westward to D

♦

Recent changes in the branch personnel of Owens-C Fiberglas Corp. have been announced. Previously m of the Cleveland office, W. H. Atkinson has succeeded L. Myers as Chicago branch manager at 3206 Pa building. Mr. Myers has joined the general sales or tion and will be engaged on special assignments. Al nounced is the return of Earl Swaim to the Toledo g offices where he will be associated with G. E. Gregory, president in charge of commercial development.

♦

In order to meet increased wartime need for critical materials, Formica Insulation Co. has added factory involving three floors of the plant at 4620 Spring avenue, Cincinnati.

♦

Transfer of W. M. Ballew of Kansas City, Mo., to the of southwestern sales manager has been announced b United States Rubber Co. In his new position Mr. B

ng, lightweight
trument carry-
case made from
carta "444".

ew, formable Micarta "444" makes molding of icate laminated-plastic shapes easy with low-cost ipment. Micarta "444" is supplied in flat sheets, molded and cured, ready for further molding into le, strong, permanent shapes by simply applying t and pressure. Heated sheets can be formed in pensive wood molds in an arbor press or wherever sures of approximately 100 pounds per square inch be applied. When cool, the shape is permanent. Strong, stable and light, Micarta "444" may be ded economically into sharp bends and deep s, without loss of characteristic Micarta proper- of resistance to heat, cold, humidity and chemi- Get the full story. Write for Micarta Data Book 84-A. Westinghouse Electric & Manufacturing pany, P.O. Box 868, Pittsburgh 30, Pa. J-06372

will be responsible for mechanical goods sales in bra fices located in Kansas City, Tulsa, Denver, Houston, New Orleans, Omaha and Minneapolis. Also annou the appointment of H. S. McPherson of St. Louis western sales manager of the mechanical goods divis territory will include Detroit, Cincinnati, Indianapoli cago, Milwaukee and St. Louis.

♦

Formerly advertising and sales development ma Grand Rapids, Mich., Carl R. Moss has been named manager of the St. Louis office of Haskelite Mfg. Corp

♦

According to a recent announcement by Allis-Ch Mfg. Co., U. E. Sandelin has succeeded A. J. Schm manager of the Seattle district office and also will su the Spokane branch office. Mr. Schmitz has been Pacific regional manager with headquarters at San Fra

♦

Westinghouse Electric Corp. recently appointed S. Ryan as assistant to the vice president. C. B. Di succeeded Mr. Ryan as manager of the feeder division, E. R. Perry has been named manager of the Micarta sion. Appointment of Leonard C. Blevins as sales ma of the meter division also has been announced. Succe him as watthour meter sales manager is H. L. Buechne

♦

Promotion of Howard Holmes to sales manager and Newcomb to sales promotion and advertising manage been announced recently by Simmonds Aerocessories In

♦

J. J. Roessle, formerly associated with Mesta Machine has joined the Pittsburgh divisional sales office of the Bearings Division, General Motors Corp.

♦

Recently made known by Bliss & Laughlin Inc., H Ill., is the retirement of L. E. Meidinger who had been trict manager at Milwaukee for the past thirty-one R. L. Mitenbuler will direct sales in the Wisconsin tory. He will be located at Room 505, First Wisconsin building, 743 North Water street, Milwaukee.

♦

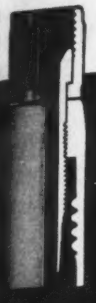
Appointment of A. E. Hess as manager of the Ho branch office has been announced by General Controls Glendale, Calif. Mr. Hess will serve users of automatic trols throughout Southern Texas, Louisiana and Sou Mississippi.

♦

Leave of absence to serve in an overseas capacity the U. S. War Department has been granted W. E. Mulles assistant sales manager for Lukenweld Inc., a division Lukens Steel Co.

♦

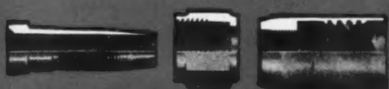
Addition of R. F. O'Brien to the technical staff has announced recently by Automatic Temperature Control Inc., Philadelphia. Previously development engineer for S Corp. of America, Mr. O'Brien now will act in the capa of instrument engineer to handle special sales and engi ing applications.



TWO MAJOR WAR DEVELOPMENTS

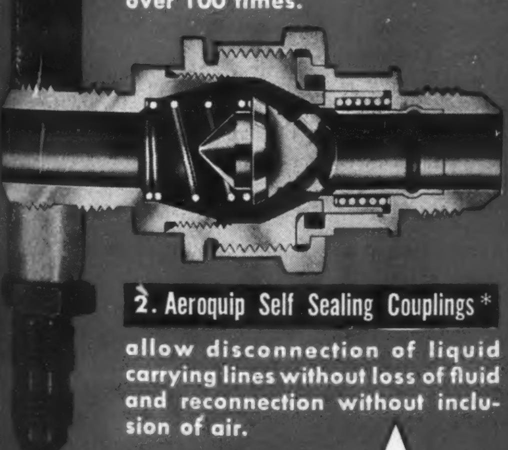
now ready for industry

1. Aeroquip Hose Lines* with detachable and reusable fittings simplify the supply problem and save valuable time, thus helping our armed forces on all fronts.



3 PIECES (each replaceable)

Assembly without special tools. No tightening or adjustment after assembly. Fittings can be removed from hose and reused over 100 times.



2. Aeroquip Self Sealing Couplings* allow disconnection of liquid carrying lines without loss of fluid and reconnection without inclusion of air.

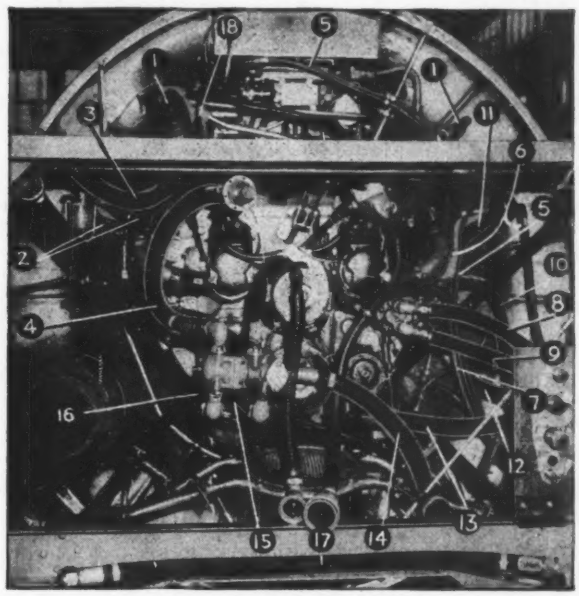
*STANDARD EQUIPMENT ON U. S. ARMY AND NAVY COMBAT AIRCRAFT

AEROQUIP CORPORATION
JACKSON, MICHIGAN, U. S. A.

5,061 HOURS OPERATION ON NEW BRANIFF INSTALLATION

Engine-change time shortened 30 minutes . . . 40 man-hours saved per overhaul.

With a 5,061-hour operational record revealing new gains in time saved and fire hazards reduced, installation of Aeroquip Flexible Hose Lines in place of rigid power plant plumbing lines on Douglas DC-3's by Braniff Airways marks a forward step in maintenance efficiency.



1.	*Oil Vent Lines*	*10.*	*Fuel Cross Feed*
2.	*Dehydrator*	*11.*	*Fuel Line*
3.	*Prop Feathering*	*12.*	*Fuel Suction*
4.	*Oil Suction.*	*13.*	*Vac. Suction*
5.	*Carb. De-icer*	*14.*	*Oil Line to Cooler*
6.	*Oil Pressure*	*15.*	*Oil Line Out from Engine*
7.	*Manifold Pressure*	*16.*	*Oil Line Into Engine*
8.	*Hydraulic Suction*	*17.*	*Oil Line Return to Tank from Cooler*
9.	*Hydraulic Pressure*	*18.*	*Primer Line*

Braniff officials report engine-change time reduced by a half-hour, and 40 man-hours saved per engine overhaul, with Aeroquip Hose Assemblies and Self-Sealing Couplings. Further time saving is reported on regular service checks due to cleanliness of power plant sections, absence of leaks minimizing wash-down work.

SAFETY IMPROVED

By elimination of 50% of the pipe joints, and with Aeroquip's leak-proof performance stopping drips in the engine nacelles, fire hazards are measurably cut. Aeroquip's assemblies meet CAA requirements as to sufficient fire resistance. The flexibility also ends problems of rigid lines under vibration.

STOCK SIMPLIFIED

With Aeroquip's quickly removable, reusable and interchangeable fittings, all parts can be replaced individually and new hose lines cut to any lengths and assembled on the spot without special tools. This feature has proved of great value in military use, where these lines are "AN" standard.

WEIGHT SAVED

Aeroquip supplies all these advantages at no sacrifice in weight; in fact, savings totalled 4 lbs. 9 oz. and 4 lbs. 11 oz. for left and right engines.

NEW MACHINES—
And the Companies Behind Them

Armament

°Airport fire truck, Cardox Corp., Chicago.
°Diesel-hydraulic fork lift truck, Whiting Corp., Harvey, Il
°Four-Forty-Four truck tractor, White Motor Co., Cleveland 1.
°M-4 tankdozer, La Plant Choate Mfg. Co. Inc., Cedar Rapids, Ia.
°M-8 light armored car, Ford Motor Co., Dearborn, Mich.
°M-1 heavy wrecking truck, Ward La France Truck Di Great American Industries Inc., Elmira, N. Y.
°M-18 tank destroyer, Buick Motor Co., Flint, Mich.
°M-24 combat tank, The Heil Co., Milwaukee 1.
°M-29C "Water Weasel", The Studebaker Corp., South Be 27, Ind.
°Scout car, White Motor Co., Cleveland 1.
°Ten-Ton "Six by Four" cargo truck, White Motor Co., Cleveland 1.

Industrial

Unit type dust collector, Ideal Commutator Dresser Co., Sycamore, Ill.
Exhaust heat recovery silencer for diesel engines, Engineering Specialties Co., Inc., New York 7.

Metalworking

Internal and surface grinder, Lempco Products Inc., Bedford Ohio.
Special milling machine with automatic electronically controlled feed rate, The Sundstrand Machine Tool Co., Rockford, Ill.
Precision belt grinder, Stuart Industries Inc., Newton 58, Ma
25-ton self-contained hydraulic press, The Watson Stillma Co., Roselle, N. J.
2-housing press brake, Cincinnati Shaper Co., Cincinnati.
Open-end bar shear, Thomas Machine Mfg. Co., Pittsburgh 2
All-purpose bench press, Maxant Button & Supply Co., Chicago 7.
"Packaged" cutting oil cooling unit, The Airtemp Div., Chrysler Corp., Dayton.
Special profile millers, Snyder Tool & Engrg. Co., Detroit
Machine for double flaring or lapping of tubing, Leonar Precision Products Co., Garden Grove, Calif.
Hydraulic stretch-leveling table, Hufford Machine Works Inc Redondo Beach, Calif.

Rubber

Ball bearing mounted expander, Mount Hope Machinery Co Taunton, Mass.

Shoemaking

Boot resoling unit, Union Supply Co., Denver 2, Colo.
Machine for pressing and staking the wedge pins in track shoes, Design & Build Div., Hydraulic Machinery Inc Dearborn, Mich.

Textile

High-speed evaporator for drying solvents, Industrial Ove Engrg. Co., Cleveland.
Dehumidifier, General Air Conditioning Co., Oakley, Calif.

Welding

3-phase resistance welder, Sciaky Bros., Chicago 38.
Spot welder, Thomson-Gibb Electric Welding Co., Lynn, Mass
Bench type spot welders, The Interstate Machinery Co. Inc Chicago 9.
Powered welding positioner, Standard Machinery Co., Providence 7, R. I.

° Illustrated on Pages 136-139.